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A STUDY OF DIFFUSION IN REVERBERATION CHAMBERS

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A STUDY OF DIFFUSION IN REVERBERATION CHAMBERS

av

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A STUDY OF DIFFUSION IN REVERBERATION CHAMBERS

KAJ BODLUND

TO

MAX AND MATTIAS

My sons

whose beloved existance

reminds me about the superior brilliance of nature



PREFACE

This report gives in its first part a summary sketch of the thesis: "A Study of Diffusion in Reverberation Chambers". In its second part the last paper of the thesis is presented.

The very mode of presentation has been chosen because the completing paper has been submitted to the Journal of Sound and Vibration for publication. However, it has not yet been accepted. (Vectors, which are to be printed in bold face, are indicated by underlining with a wavy line.)

The thesis comprises the following papers:

1. K. BODLUND 1976 Journal of Sound and Vibration 44, 191-207.

A new quantity for comparative measurements concerning the diffusion of stationary sound fields.

2. K. BODLUND 1976 Journal of Sound and Vibration 45, 539-557.

Statistical characteristics of some standard reverberant sound field measurements.

3. K. BODLUND 1976 Division of Building Technology, Lund Institute of Technology Report 74 (Part 2). A study of diffusion in reverberation chambers provided with special devices.

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PART 1

A SUMMARY SKETCH OF THE THESIS:

A STUDY OF DIFFUSION IN REVERBERATION CHAMBERS

1. BACKGROUND

Most of the standardized acoustic reverberation chamber measurement methods are based on the condition that the generated sound fields are diffuse. This statement is applicable to the measurement procedures concerned with the determination of e.g. sound power levels emitted by small noise sources and airborne sound transmission loss of partitions. A sound field being diffuse implies

- a) that the sound pressure level fluctuations are small within the chamber (easy to obtain accurate measurement values)
- b) that we have a large number of individual sound waves, the frequencies and incidence angles of which are all uniformly distributed (giving representative measurement values)
- c) that with the quantity sound pressure level (a quantity that is measurementally easy to estimate) it is possible to identify values of some interesting quantities that are difficult to measure directly, e.g., sound power, sound transmission coefficient and sound absorption coefficient.

The validity of this condition has been penetrated in literature for several different situations. Empirically we know that the sound fields are far from perfectly diffuse at low frequencies, when the wavelengths are comparable with the room dimensions. Several different measures have been proposed and applied for investigations about the sound field diffusion, and various more or less adequate treatments have been supplied to the empty reverberant chamber in order to enhance the diffusion at low frequencies. However, most of the associated measures of diffusion have not been any great success. Consequently our knowledge about the effects attained by the different treatments are limited,

especially when 1/3-octave band stationary noise is used for generation of the sound fields.

To complete the background of this work, it should be pointed out that severe non-repeatability has been observed when Round Robin tests of the transmission loss, sound power and absorption coefficient measurement procedures have been performed. These experiments indicate that a possible source of error at low frequencies is the lack of diffusion. From a logical point of view, the importance of this source of error should have been accentuated by the fact that the various laboratories involved all had a different geometrical design and that different additional so called diffusing treatments were used.

2. PURPOSE

The presented research work has been made for the purpose of getting a description of the problem: unperfectly diffuse stationary sound fields. Consequently the work has been concentrated upon finding answers to such questions as,

- * how shall an adequate and sufficiently effective quantity for measurements of sound field diffusion be constructed?
- * in what state of diffusion do we find the normal empty reverberation chamber sound fields (regarding e.g., such factors as frequency and room volume)?
- * how is the diffusion affected by various treatments?
- * how do the associated alterations affect the output of our standardized measurements?

However, because of practical reasons it has been necessary to restrict

the investigations accordingly,

- * only normal (100-200 m³) hard-walled laboratory chambers have been studied
- * only stationary 1/3-octave band white noise excitation has been studied
- * the following chamber alterations have been studied: various source arrangements, additional absorbents, stationary hanging reflectors and moveable reflectors
- * the practical consequences of the various alterations have been checked by sound pressure level difference measurements only
- * the various chamber treatments have only been investigated in a surveyable manner, concerning the parameters characterizing each specific treatment.

3. METHODS

The first question presented in section 2 above has been penetrated in paper 1. With the aid of literature experiences and some experiments of my own, a new quantity suitable for comparative measurements of sound field diffusion has been constructed. In paper 1 and 3 this quantity has proved to behave logically in comparison with existing theories and empirical knowledge. Furthermore an excellent repeatability has been observed for these diffusional measurements.

As indicated above it is necessary to perform some comparative standard measurements too, so as to check the practical consequences of insufficient diffusivity in connection with alterations studied. When doing so one must know the measurement accuracy, to be able to

evaluate if the observed differences are significant or not. When reading the available literature, it was found that further research was needed within this area. Such a work was therefore carried out, and the results are presented in paper 2. Accordingly it has become possible to estimate, in a simple way, confidence intervals for the standard measurements of sound pressure level, reverberation time, sound pressure level difference, sound power level and transmission loss.

The two measuremental instruments presented (paper 1 and 2) form the basis of the comparative test procedure, which has been applied on various situations of interest in paper 3. The two ways of measuring have been used in a parallel manner, in order to obtain a qualitative evaluation of the various chamber treatments.

4. RESULTS

Throughout this work the experimental results have fully confirmed that the empty chamber sound fields are far from being perfectly diffuse at low frequencies. This circumstance is accompanied by the fact that the mean square pressure spatial variance is larger the lower the frequency is. However, by using an adequately designed mean square pressure sampling procedure, it has been possible to estimate and maintain the measurement accuracy at low frequencies (regarding the time and spatial fluctuations only).

In agreement with existing literature experiences it was observed that the choice of source arrangement was important to the sound field diffusion; the use of absorbents affected the diffusion negatively at high frequencies; stationary hanging reflectors improved the diffusion of the empty reverberation chamber only in a small chamber and for the

highest frequency band investigated (1250 Hz).

Before the intended diffusional moveable reflector experiments were carried out, some guiding experiments were performed in a model reverberation chamber. This course of action was chosen because, although the moveable reflector is generally regarded as the most effective device in enhancing diffusion, there has not been found any detailed recommendations about how to design such a reflector. Besides, several practical problems are associated with the realization of such a diffuser. The conclusion made from these model experiments was that an adequately designed moveable boundary system has considerable advantages compared with the traditional rotating diffuser. Consequently a full-scale moveable system was realized, and the intended experiments were carried through. By performing a room averaging with this specially designed discretely moveable boundary system, drastical improvements were observed at low frequencies for both the diffusivity and the mean square pressure spatial variance.

When performing the respective sound pressure level difference measurements it was observed that all the treatments studied had some practical importance. This implies that the way of using these treatments should be the subject of specific recommendations in measurement standards.

PART 2

A STUDY OF DIFFUSION IN REVERBERATION CHAMBERS
PROVIDED WITH SPECIAL DEVICES

SUMMARY

The main purpose of this paper is to penetrate in a surveyable manner some of the most common and important ways to achieve sound field diffusion. This purpose has been effected by the use of a specially designed, experimental and comparative test procedure. The basis of this procedure is a measure which quantitatively estimates the degree of diffusion. When comparing the outcome of the experiments with applicable theories and literature experiences, the measure has proved to behave logically.

The use of more or less complex loudspeaker arrangements, additional patches of absorbents, stationary hanging reflectors and moveable panels have been studied. The extreme results can be summarized accordingly: An increasing degeneration of the diffusion was obtained at high frequencies when absorbents were added to the empty reverberation chamber; extraordinary but statistically expectable improvements were attained by the use of a specially constructed, so called moveable boundary system. This system is functioning according to the same idea as a rotating diffuser. However, the reflective panel locations are discretely altered and the total one-sided reflective area is comparatively large (58 m^2). Because of the use of such a displacement system, it has been possible to outline a procedure with which to search for effective combinations of boundary configurations and evaluate the effectiveness of the associated room averaging. Applying the indicated selection procedure on the 125 Hz situation, it was possible to find a suitable set of five boundary configurations. Performing the associated room averaging, the spatial variance of the mean square pressure decreased by a factor of 0.14-0.44 and the diffusivity values decreased to a level typical for the empty chamber when 500 Hz excitation was used.

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The various devices investigated have also proved to affect significantly the outcome of some standard transmission loss measurements. This fact points out the practical consequences and indicates that these matters should be considered at the development of future standards. Although a transmission loss laboratory has been used to check the practical consequences, the discussions in this paper are also applicable to other standard methods, e.g., to the methods for determination of sound power levels.

1. INTRODUCTION

When performing a standardized reverberation chamber measurement we have to make the basic assumption that the sound fields are sufficiently diffuse, see e.g., the references [1-3]. Accordingly, it is of vital importance to be able to describe and estimate the sound field diffusion. Otherwise we are running the risk of getting incorrect estimates when unknowingly applying the measurement procedures on situations with insufficient diffusion.

Much research work has quite naturally been focused on this subject. A wide and qualitative discussion of this research is given in reference [4]. From both theoretical and experimental observations we know that the degree of diffusion in normal empty reverberation chambers diverges considerably from perfect diffusion, especially in the low frequency region. Therefore, various special devices have been brought into our measurement procedures in order to enhance diffusion. Furthermore, different experimental methods have been developed and applied in order to evaluate the effects of these special treatments. Unfortunately, none of the methods has been very successful [4] and our knowledge about the importance of the various devices commonly used is relatively limited. Especially, we do not know very much about the low frequency diffusion in reverberation rooms excited with stationary 1/3-octave band noise signals. One reason is that most of the experimental research work performed has been devoted to reverberation time measurements and to chambers equipped with a large and concentrated sample of an absorbent. The statements above are elucidated by the literature survey given in section 2 below. Furthermore, we know that severe non-repeatability has been observed in the standard

1/3-octave band measurements, see e.g., the references [5-6]. In reference [5] a Round Robin test of the transmission loss measurement procedure is reported. The measured values vary considerably between the different laboratories. In reference [6] some comparative measurements of sound insulation, made by several different organizations in one and the same laboratory house, are reported. Also in this investigation considerable variations are observed. The complete system of potentially influencing parameters in connection with an investigation like those in reference [5] or [6] is somewhat intricate. E.g., at the comparisons made in reference [5] several of the possibly important parameters are changed simultaneously. As the investigations referred to have been limited only to deal with some of the conceivable sources of variations, it is quite natural that it has not been possible to explain all the variations observed.

With this background it is highly motivated also to find an answer to the question: What rôle plays the basic assumption that the stationary sound fields are diffuse for the laboratory reverberation room measurements? A special experimental test procedure outlined in the introduction of reference [7] has been developed in order to search for an answer to this question. The method, which will be discussed in section 3 below, has been applied on several situations of interest, and the main subject of this paper is to present and discuss the most important results of these experiments.

For different reasons the variations observed in [5] and [6] are not restricted to low frequencies only. Because of this typical observation and in order to obtain also a frequency survey of the sound field characteristics, the investigation is performed

for the typical low and midfrequency 1/3-octave bands: 125, 250, 500 and 1250 Hz. Furthermore, it is believed that it is not practicable to achieve diffusion for the typical field situation at low frequencies. So this investigation is limited to deal with laboratory conditions. Another important reason not to investigate the field transmission loss measurement situation is indicated by reference [8] and [9]. Schultz shows that a test procedure involving A-level differences is just as useful and successful as the more elaborate and expensive methods involving 1/3-octave band data, when predicting achieved privacy in dwellings. When studying A-level differences the sound field diffusion is an irrelevant factor.

2. A LITERATURE SURVEY OF HOW SPECIAL TREATMENTS AFFECT THE DIFFUSION

The investigations referred to below are concerned with sound fields in reverberant laboratory chambers. Most of them also study the decaying sound field after the sound source is switched off. Although this paper is basically a study of stationary fields, conclusions drawn in these investigations are interesting, as we cannot reject their applicability.

In theoretical investigations concerned with reverberant sound fields the sound source is mostly regarded as a monopole point source. This is an idealized model and the excitational conditions have been little investigated theoretically, for ordinary loudspeakers. Little examined is also the question of how the state of diffusion is dependent on the source characteristics.

In reference [10] it is experimentally found that the directional radiation of the loudspeaker is important. Measurements of decay rates are performed on a large area of a highly absorbing sample. Observed non-linearities in the decay curves are accounted for by the fact that waves grazing the material will have more energy than the other waves, especially during the later part of the decay. This non-linearity could be suppressed at high frequencies (1000 and 2000 Hz) by turning the loudspeakers towards the ceiling. Originally the two 12-in loudspeakers used were facing the walls. This change is said to be caused by the fact that more energy is supplied to the oblique waves. The hypothetical explanation of the observed phenomenon thus presupposes and implies that the resonant waves can easily be excited to widely different levels in the steady state. At field measurements of transmission loss

it has also been observed that the choice of source type and location can be important, see e.g., reference [6]. At such measurements we must also remember that possible flanking transmission may be dependent on the parameters characterizing the source. The references mentioned give no direct answers to the question above. They only imply that the choice of sound source may be crucial.

The geometrical properties of a reverberation chamber cannot easily be modified for experiments. Of course comparisons between different chambers can be carried through, but then, unfortunately, the difference mostly lies in more than one parameter. Therefore, the importance of proper room proportions, splayed walls and room volume has been analysed mainly through theoretical discussions. In the references [11, 12] the importance of proper room proportions are theoretically investigated for rectangular rooms. At low frequencies, where there are few eigenfrequencies, the observation is made, that the frequency and angular distribution of the respective normal-modes are essentially influenced by the room geometry. Recommended room proportions are for instance 1:0.82:0.72. In reference [13] a perturbation method is demonstrated, which makes it possible to find solutions of the wave equation for room shapes characterized by "small" deviations from a basic, soluble shape. By means of qualitative discussions based on the general formulas derived, boundary irregularities are said to result in a higher state of diffusion. The more non-symmetries a room possesses, the larger will be the sound diffusion. Irregularities of the order of a half wavelength in size are recommended.

One of the most frequent attempts to achieve diffusion is to add stationary reflective elements to the room boundaries or the

space, e.g., boundary relief or hanging panels. Regarding the hard-walled rectangular chamber, it is concluded from a theoretical discussion, that the addition of scattering objects can reduce possible sound field degeneracy at low frequencies ([14] and [15] p.575). In theory the number of eigenfrequencies is not essentially altered if the chamber volume is not changed. This fact, and the circumstance that ordinary chambers seldom have such simple shapes as discussed in connection with degenerate fields, imply that the fixed diffuser does not give any positive effects in ordinary hard-walled chambers [14].

In experimental studies of the decaying sound field it was found that reflecting panels are important for the results of absorption measurements and for the degree of diffusion, see e.g., the references [10, 16, 17]. The conclusions drawn in all these investigations were that hanging panels provided diffusion in the decaying state. In reference [17], dealing with measurements with a directional microphone in a model chamber, the stationary sound field was also examined. The scale of the model experiment was chosen so as to simulate original frequencies from 800 Hz to 2 kHz and original room volumes from 300 m^3 to 3000 m^3 . (The basic problem with directional microphones, that is, the difficulty of obtaining directional sensitivity at low frequencies, can be avoided by measuring in a model scale.) These experiments show that the degree of directional diffusion increases if the panel area increases and if the floor is covered with absorbents. (The lengths of the edges of the diffusing panels were 3-8 wavelengths, the absorption coefficient of the floor absorbent $\alpha_f \approx 0.93$, the average absorption coefficient of the model chamber with floor absorbent $\alpha \approx 0.20$.) The dependence is asymptotic, and above 30 %

relative panel area (total area of both sides of the panels/total wall area) no effect is observed. If the floor is made reflecting, only a small improvement can be observed, as the relative panel area increases from 0 % to 6 %. Above this value no dependence on panel area could be found. The degree of diffusion is also essentially higher in the empty chamber than in the chamber with absorbents. The results in reference [17] also coincide with the results obtained in a full scale reverberant chamber at 250 Hz in reference [18]. When the floor was covered with a sound-absorbing material the diffusivity increased as the relative panel area was increased up to 35 %. When only a part of the floor (25 %) was covered with the same absorbent, the diffusivity increased only when the relative panel area was increased to 17 %. The diffusivity also decreased with increasing area of the absorbent. The effect of adding diffusing plates to the empty reverberation chamber is not reported in reference [18].

Thus the effectiveness of hanging panels in the steady state is well documented if the reverberant room is equipped with a concentrated sample of an absorbent. However, the effectiveness in a highly reverberant room is doubtful. Furthermore, it is well documented that the method is effective in the decaying state. However, it is important where to record the reverberation signals, see reference [10], and that the test specimen is not hidden in a forest of panels, see reference [4]. Instructions concerning the panel parameters for absorption measurements is given in reference [16]. I quote: "It is generally considered beneficial to have the panel dimensions comparable with the wavelength of sound at the lowest frequencies of interest. They can be slightly curved in shape. They should not contribute appreciable sound absorption and

should act as specular reflectors. The total area of one side of the diffusers should be approximately equal to the floor area of the room and the projection on each room surface roughly proportional to its area."

In model experiments, reference [19], the effect of attaching hemispheres to the walls was examined, on determining absorption coefficients. These experiments also included a comparison with the effect of hanging panels. The result was that hanging panels are better than the boundary relief. Using the same test method, but in full scale, the same result was obtained in reference [20]. The hypothetical explanation given to this phenomenon, reference [19] p.31, is that there exists a difference in edge absorption. If this is true, the test method, which is a comparison between the observed absorption coefficients and a theoretically derived coefficient, does not actually give us any knowledge of the difference in diffusion. Boundary relief experiments concerning the steady state have not been found. An important thing to remember is that addition of boundary relief decreases the chamber volume and consequently the number of eigenfrequencies. This especially happens when we want to get effects at low frequencies, which demand large dimensions.

In the references [4, 13], absorptive patches randomly applied to the walls are said to be less efficient than boundary relief of the same size. This statement is made on the basis of a qualitative theoretical discussion. The effect of adding large and concentrated samples of absorbents is treated in literature and in connection with the discussion above concerning diffusing panels. The effects observed have only been negative, a fact supported by the classical normal-mode theory. For both these

cases we know that, if we introduce too much absorption, the sound fields will become semi-reverberant, that is, the importance of the direct field of the loudspeaker will increase and, consequently, the sound field diffusion decreases.

A carefully designed moving diffuser is generally regarded as the most effective device in enhancing diffusion. Hypothetically the function of the moving vane is that it continuously shifts both eigenfrequencies and incidence angles. That is, by using sufficiently long integration times, a room-averaging effect will be obtained hypothetically.

By use of radiation resistance measurements the effects of moving vanes are investigated for low frequencies in reference [21]. Through power measurements and other experiments we know that the sound power output in a reverberant chamber is dependent on source location. The power output from a source is dependent on the radiation resistance presented to it. This resistance and its dependence on location and frequency is theoretically determined by the modal structure. This dependence can be derived for an empty, rectangular chamber, see e.g., reference [21], equation 5. If a loudspeaker is moved in the room, the source coupling to the modes and the power output are changed, but not the modal frequencies. However, alteration of the boundary conditions causes a change in eigenfrequencies and in power output. Comparative experiments showed that the variations in radiation resistance were larger when the source position was varied than during measurements with a rotating diffuser. However, the variations observed during the rotations signify that it is possible to average out some of the dependence on chamber and source location at power measurements. By sweeping the source frequency and letting the diffuser be

fixed at different angles, different power outputs were measured. The output power frequency curves were changed and an observed peak was shifted (1.5 Hz) as the diffuser was differently oriented. The explanation of this result is that the changed boundary conditions cause an alteration of the resonance frequencies and thereby a change in power flow. However, the variation in power output caused by the rotations of the vane is smaller than the variations observed when sweeping the source frequency. Concerning the construction of the rotating diffuser, see reference [21], p.109. The result of the investigation above raises the natural question, whether it is possible to make the observed shifting in resonance frequencies more effective, and consequently to improve the sound field diffusion by using a somehow different moving diffuser.

In reference [22], where a cross-correlation technique is used, two loudspeakers mounted on large rotational vanes are studied, that is, a combination of a continuous change of source locations and boundary conditions. The main purpose of the investigation was to check if the measured values of the correlation function agreed with the results predicted for a diffuse sound field. Close agreements were observed when using the rotating vanes. Furthermore, I quote from the conclusive section p.1077: "It was demonstrated that this agreement was not obtained if the rotation of the vanes was absent." This statement is not fully confirmed by a direct measuremental comparison in the reference.

An investigation dealing with the effects of rotating diffusers on the spatial variance of the mean square pressure is presented in reference [23]. The experiments were performed in a

model chamber, and the results show that the spatial variance for pure-tone sound fields could be reduced by a factor up to 5, depending on the frequency and the design of the diffuser. No specific recommendation of a suitable diffuser shape is given, except that it should be as large as possible.

Although much research has been devoted to the moving diffuser, it is pointed out in literature, e.g., in the references [21, 23], that further research has to be made. This necessary research work should be concentrated on obtaining a theoretical justification for the use of the rotational diffuser and on finding optimum diffuser parameters. The recommendations of suitable diffuser designs available in literature, reference [4], can be summarized by the following points:

- (i) Dimensions of the same magnitude as the wavelength of sound at the lowest frequencies of interest.
- (ii) A reflecting surface, that is, the surface must weigh at least 4.9 kg/m^2 if 90 Hz is the lowest frequency of interest.
- (iii) A suitable form, so as to affect all room modes.

3. EXPERIMENTAL TEST PROCEDURE

3.1. TEST PHILOSOPHY

As a foundation for an efficient test method it is necessary to choose a measurement quantity that estimates the degree of diffusion and that makes comparisons possible. This choice is discussed in section 3.2 below. Furthermore, to evaluate quantitatively the practical importance of variations in diffusion and the practical consequences of the respective treatments investigated, the comparative procedure must include some standard measurements. As the diffusion quantity is only developed for measurements in stationary sound fields and, therefore, we are bound to deal with stationary signals, the logical choice is made, to use sound pressure level difference experiments for this purpose. This quantity is fundamental for the measurement of airborne sound transmission. The accurate performance of such comparative level difference measurements is discussed in section 3.3 below.

For the different comparable cases studied, the test method involves a comparison between measured degrees of diffusion and the respective measured values of the primarily interesting factor, the sound pressure level difference (denoted D) for a particular wall. If, in this respect, there is a statistically significant difference between values of D and at the same time a statistically significant difference in the measured degrees of diffusion, we cannot reject the hypothesis that there is an essential difference in diffusion between the studied sound fields. If we get a significant difference in D -values but no difference in degrees of diffusion, it is logical to treat the different measurement

results as samples of one and the same population. If, on the other hand, we get a difference between degrees of diffusion but no difference between measurement results, the changes that cause an increase in diffusion are naturally of no use for the particular wall studied. In the last-mentioned case it is logical to examine the behaviour of some more walls. If, by doing so, one or some new objects give a significant divergence, one cannot reject the hypothesis that, for laboratory measurements, a substantial increase in diffusion has been obtained.

For practical reasons, but also to check the generality of the results of the diffusion measurements, three different reverberation chambers are used. Chamber 1 is an almost cubical rectangular room with the proportions 1:0.93:0.92 and the volume 190 m^3 . The geometry of chamber 2 is shown in reference [7], Figure 7, and its volume is 104 m^3 . Chamber 3 is the source room of the transmission loss laboratory, which is presented in reference [5], Figure 9, and its volume is 132 m^3 . Concerning the experiments in the transmission loss laboratory, the diffusing devices studied have been inserted into the source room (chamber 3) and the receiving room condition has been kept constant. Accordingly, the two chambers have always been unequally equipped and the receiver room has been provided with a normal amount of randomly hanging panels.

3.2. METHOD FOR TEST OF SOUND FIELD DIFFUSION

The developed test procedure is based on a new quantity for comparative measurements of diffusion. This quantity is denoted ϵ_d and its fundamental characteristics are discussed in reference [7] and below. Two dynamic pressure microphones (BEYER type: M101N) and a digital correlator (SAICOR type: Sai-42) are used. Furthermore, a special microphone stand is constructed with which one of the microphones is discretely movable on a sphere with a prescribed radius, while the other microphone is fixed at the centre of the sphere. The microphone stand can be moved around, so that the "centre" microphone can be placed at random positions within the interesting reverberant sound fields (avoiding the boundaries and the free fields of sources).

The specific quantity ϵ_d is basically a typical spatial variance of the cross-correlation function for the sound pressures from the microphones. Therefore, to obtain a sample of the quantity ϵ_d , with the experimental arrangement above, cross-correlation function measurements are performed in 6 randomly chosen directions around the "centre" microphone. The spatial variance is estimated by using a theoretically derived ideally diffuse cross-correlation function as the expected function. Consequently, the lower values we obtain for ϵ_d , the more diffuse is the sound field. Further important properties of ϵ_d have been experimentally investigated in [7] and the quantity has been observed as being sensitive both to directional and frequency composition of complex sound fields. For further details concerning ϵ_d , see reference [7].

The situations studied in this paper are usually investigated by taking at least four independent spatial samples of ϵ_d . The various common

devices of interest (e.g., randomly hanging panels) are examined separately and comparisons are made between the results obtained for these cases and specific reference cases (e.g., the empty room case). As indicated above and because of the fact that ϵ_d is a random variable, it is necessary to execute a statistical test to be able to conclude whether we have obtained a positive treatment effect or not. The non-parametric Wilcoxon, Mann-Whitney test discussed in reference [24], p.135, is suitable for this purpose, as the basic statistical distribution of ϵ_d is unknown. When performing such a test the ϵ_d -observations from the two comparable cases should be ranked and the sum of the ranks ΣR for the smallest set should be derived. Generally throughout this paper the observation with the highest ϵ_d -value is given the lowest rank (1). Using such information, the hypothesis that we have obtained no effect is rejected at the 0.05 level of significance, if $\Sigma R \leq x_i$ (lower tail) or $\Sigma R \geq x_j$ (upper tail). Appropriate values of x as a function of the number of observations are available in reference [24], table B. E.g., if the sum of the ranks equals 11 for a situation with 4 observations of ϵ_d from each of two comparable cases, then the hypothesis above can be rejected.

3.3. METHOD FOR ESTIMATION OF THE PRACTICAL IMPORTANCE OF TREATMENTS

When carrying out level difference measurements that shall be compared with each other, it is necessary to keep all secondary factors constant. For this reason only one frequency band at a time is studied, and the temperature and humidity are kept constant. Furthermore, the stationarity of the system is controlled by using a reference position in the source room.

As we are only interested in comparisons, most of the possible systematical errors are unimportant. However, there is one source of error that might be important. The attenuator setting of the microphone amplifier must be changed when moving the microphone from source to receiver room. Thus, for a measurement of D , we have a pair of attenuator settings, which shall be used in the computations of D . Then if we use different pairs of settings when making a comparison between two cases, we run the risk of introducing a systematical difference. This risk can be easily eliminated by keeping the attenuator settings constant during comparative measurements and instead control the accurate working of the instruments by the output level of the sound source.

The measurements of D presented in this paper have been carried out according to the recommendations in reference [1] and in an ordinary transmission loss laboratory. For geometrical details of the laboratory, see section 3.1 above. Discrete spatial samples (20 samples for 125 and 250 Hz, 10 samples for 500 and 1250 Hz) are randomly taken in both source and receiver room, and an integration time of 1 minute is used throughout the paper. With the method described above it has been possible

to study up to 5 different source room treatments for a single frequency band during one normal working-day.

When comparing various D-values, as well as when comparing the respective ϵ_d -values, it is necessary to make a statistical consideration about the results. The basic statistical characteristics of the random variable D are discussed in reference [25]. According to the principles in this reference, it is possible to estimate 95 % confidence intervals for D. These approximate confidence intervals are derived, and the conclusion is drawn that a significant and important change has been obtained, if two comparable cases have no overlapping confidence intervals.

To check the evaluating method, three measurements have been performed at one and the same source room condition. The results demonstrated in Figure 1 are fully positive.

FIGURE 1

4. DIFFERENT STATIONARY SOUND SOURCES IN THE REVERBERATION CHAMBER

4.1. BASIC THEORY

To present a basis for the low frequency experiments, the theoretical relations below are included and discussed.

By using complex notation the time-averaged sound intensity $\underline{I}$ radiating from a harmonic source can be written as

$$\underline{I} = \frac{1}{2} \operatorname{Re}(p \cdot \underline{u}^*) \quad (1)$$

where p is the pressure and $\underline{u}$ the particle velocity of the sound field. The asterisk denotes the complex conjugate and Re indicates that the real part is to be studied. If the intensity varies around the source the total power output W will be found by integrating over a convenient surface surrounding the source

$$W = \iint \frac{1}{2} \operatorname{Re}(p \cdot \underline{u}^*) \, ds \quad (2)$$

By inserting the radiation impedance Z_r (Ns/m^3), that is, the ratio between the sound pressure and the acoustic particle velocity normal to the surface, the following wellknown expression will be obtained

$$W = \iint \frac{1}{2} |\underline{u}|^2 \operatorname{Re}(Z_r) \, ds \quad (3)$$

where $|\underline{u}|$ denotes the magnitude of $\underline{u}$.

As we are basically interested in having a smooth characteristic of the source output, it is now convenient to study the variables in equation (3) versus frequency. First we know that in normal

reflective chambers the radiation impedance depends on the source pressure, the reflected pressure, the source particle velocity and the reflected particle velocity. This is further discussed in the references [26, 27]. Therefore the radiation impedance varies considerably at low frequencies in hard-walled chambers. Ebbing has studied this phenomenon both experimentally and theoretically in reference [21].

Choosing the most suitable integration surface - the vibrating surface of the source - the other variable of interest is the source vibration velocity. Furthermore, we make the simplifying assumption that we can regard the source as a small piston source, with an effective area S vibrating with a uniform velocity $\underline{u}$ normal to the surface. This velocity depends on the force driving the source, the radiation and the internal loads. The internal load is expressed by the aid of the internal impedance Z_i (Ns/m^3), see e.g., reference [28]. By using the variables above, the following relation will be obtained between velocity and driving force F

$$\underline{u} = F / (Z_r + Z_i) S \quad (4)$$

Insertion of equation (4) into equation (3) will give

$$W = \frac{1}{2} |F|^2 \text{Re}(Z_r) / |Z_r + Z_i|^2 S \quad (5)$$

From equation (5) we can see that when the internal source impedance is small compared to Z_r , then the power output will decrease if Z_r increases. Such a source is called a low impedance source. If instead the internal impedance is large compared to the radiation impedance, then the power output will increase if Z_r increases. Such a source is called a high impedance source. As reported by Ebbing and others, the radiation impedance varies considerably with

frequency at low frequencies in hard-walled chambers. From the discussion of equation (5) it is quite clear that such small-scale frequency variations cannot easily be suppressed in order to obtain small variations in power output, by the use of a single source type. The reason for this is that for normal sources the Z_i -dependence on frequency is significantly slower than the respective Z_r small-scale frequency variations. Instead equation (5) implies that the power output variations can possibly be suppressed by the use of a proper combination of a low impedance and a high impedance source driven simultaneously.

The discussion above gives an indication, how to search for an optimum excitation system at low frequencies. However, such an investigation regarding all the variables involved in equation (5), must necessarily become quite extensive and fall beyond the scope of this paper. Therefore only some sound sources, typical for the standard measurement procedures, are investigated below. The respective internal and radiation impedances are not studied quantitatively.

4.2. EXPERIMENTS

1/3-octave band filtered white noise is used in all the measurements and during the diffusion measurements the frequency characteristics of the output signals are controlled so as to avoid loudspeaker distortion. All the chambers used for the diffusion experiments in this paragraph have been empty, except for the source and the microphone equipment.

The factors primarily characterizing the excitation of the sound field are source type and location and these factors are investigated. The number of sources has also been varied.

By use of the modal theory it is often assumed that the most uniform excitation of the sound field is obtained if a point source is placed in a corner. Furthermore, we know that, at least for closed-back loudspeakers, if the dimensions of the loudspeakers are small compared with the respective wavelengths, and if they are located close to the reflective surfaces, then the direct radiation is approximately non-directional. To check the reference source (a Goodman Axiom 301 loudspeaker mounted in a $0.44 \times 0.44 \times 0.28 \text{ m}^3$ closed cabinet) with these facts in mind, some small loudspeakers were examined. The loudspeakers were placed as far as possible into one and the same corner. The result of the experiment is demonstrated in Figure 2. No significant effect on ϵ_d is obtained.

FIGURE 2

Further experiments concerning the location of single sources have been performed. None of these tests has shown that ϵ_d significantly depends on the location of the source. Only source locations close

to the walls have been examined.

In reference [6] it was observed that the spatial uniformity of sound pressure levels increased if the back panel of the speaker was removed. This result was obtained in a source room at 125 Hz. In the receiver room no dependence on source type was observed. The importance of loudspeaker size, cabinet/baffle design, baffle orientation and signal level was also examined. The first modification above gave the most significant effect among all the experiments performed. When this experiment was repeated the results in Table 1 were obtained. A slight change occurred. When using the non-parametric Wilcoxon, Mann-Whitney test, the hypothesis can be rejected, that no positive effect has been obtained at the 0.001 level of significance. So, both the results in reference [6] and this last experiment imply that the use of an open-back cabinet increases the degree of diffusion at low frequencies.

TABLE 1

In the next experiment a quite different type of loudspeaker is used. This loudspeaker box is larger and has an "Acoustic Resistance Unit" (this opening is placed 0.54 m below the centre of the speaker cone). It is constructed to be a good bass-loudspeaker. Furthermore, the cabinet is built to fit in a corner with the speaker axis horizontal and 45° to walls. The measurements of ϵ_d were performed in both chamber 1 and the source room of the transmission loss laboratory (chamber 3). The result demonstrated in Figure 3 shows a mixture in response. Significant improvements are observed in both chambers at 125 Hz and in chamber 3 at 250 Hz.

FIGURE 3

We have observed that ϵ_d is sensitive to the choice of loudspeaker construction, although our investigation has been restricted to some ordinary moving coil speakers. According to the discussion in section 4.1 above, this fact indicates that the source velocity is not independent of the radiation impedance variations for all the specific loudspeakers investigated. On the contrary, in most of the investigations reported in literature concerned with the coupling between source and reverberant sound fields, the source is assumed to be of the high impedance type. That is, the source velocity is regarded as unaffected by the acoustic environment in which the source is located, see e.g., the references [21, 26, 29].

To check the statement above, some measurements were performed with a velocity microphone (STC type 4115) close to the source surfaces of the reference speaker and the bass-loudspeaker (source No. 2 in Figure 3). Slowly sweeping pure-tone excitation with frequencies within the 125 Hz 1/3-octave band was used. It was observed that the reference source works as a high impedance source, because the same smooth velocity response was obtained both in an anechoic chamber and in the reverberation chamber 1. However, for both the loudspeaker opening and the A.R.U.-opening of the bass-loudspeaker, the velocity responses changed when this source was moved from the anechoic to the reverberant chamber. The sound velocity level curves obtained in the reverberant chamber fluctuated around the respective curves obtained in the anechoic chamber (typical deviations from the smooth anechoic chamber velocity curves were 1.5 dB for the loudspeaker opening and 2.5 dB for the A.R.U.-opening). So, the bass-loudspeaker investigated is

an example of a source that is not of the high impedance type. To get a further evaluation of what has happened, the mean-square pressure frequency responses of the respective sound fields are measured and the experimental results are given in Figure 4. Significant differences between the curves have been obtained. The one representing the best of the two sources regarding ϵ_d has a considerably smoother frequency response. The peak frequencies for the two cases are almost the same and the number of peaks is not essentially altered. The effect demonstrated in Figure 4 is logical in comparison with the results of the source velocity measurements. If the internal impedance of a source is appropriately decreased from being large in comparison with the radiation impedance, then the power output dependence on the radiation impedance variations will also decrease and a smoother pressure response curve will be obtained (see equation (5)). The logical relations observed above strongly indicate that a higher degree of diffusion has been obtained at 125 Hz.

FIGURE 4

The experiments with more than one loudspeaker started with measurements in chamber 1. Throughout these experiments the loudspeakers simultaneously used were excited with uncorrelated signals. As could be expected, all the two-source experiments did not give significant improvements. E.g., when two Goodman Maxim loudspeakers were used for some 250 and 500 Hz experiments, no significant diffusivity improvements were observed compared with the single loudspeaker cases, although the speakers were placed in corners and at edges as well as in the free space, and several

various speaker axis directions were used. However, by exchanging some of the sources primarily used, the successful composition presented in Figure 5 was obtained. Compared with the best single loudspeaker cases, significantly lower diffusivity values were obtained, except for 250 Hz, where the use of a single Goodman Maxim gave the same results as the use of two speakers did. Exchanging the 250 Hz speakers, this experiment was repeated in chamber 3. The results presented in Figure 6 show that the specific loudspeaker composition investigated is just as successful in chamber 3 as it was in chamber 1. Further experiments with more than two loudspeakers gave no significant effects. The results are logical, as an increase in the number of sound sources has never resulted in an increase in ϵ_d -values.

FIGURE 5

FIGURE 6

Studying the improvements obtained, a final comment can be added to the discussion of the ϵ_d -experiments. Measured ϵ_d -values for the reference source in the empty reverberation chambers 1 and 2 are presented in reference [7], Figure 8. If we study this figure and the differences above, we can see that the largest improvements to some extent can be explained by the fact that cases seem to exist, in which one loudspeaker in one specific chamber is more unfavourable than logically expected. Such is the case for the reference speaker at 125 Hz in chamber 1, as the associated values of ϵ_d are equal to the corresponding values in the considerably smaller chamber 2. When making experiments with another loudspeaker, values at a more logical level were obtained, see Figure 3. The same situation also

exist in chamber 3, but at 500 Hz. From the theoretical point of view, the reference values of ϵ_d (Figure 6) are too high in comparison with the values in chamber 1 and 2 and the improvements obtained when measuring with two loudspeakers are large compared with the similar cases.

The next step is to check the practical significance of the observed alterations with the sound pressure level difference procedure discussed in section 3.3 above. The experiments performed are restricted to deal with the arrangements that have given the best diffusivity values in comparison with the reference source. When measuring D with a typical partition construction mounted in the frame of the transmission loss laboratory, the results shown in Figure 7 were obtained. As can be seen, significant differences have been obtained at 250 and 1250 Hz. Some further D-experiments (at 125 and 250 Hz) with the bass-loudspeaker and another wall construction (2.5 mm laminate + 45 mm mineral wool - 180 kg/m³ + 2.5 mm laminate and timber studs connecting the surface panels with 0.4 m pitch) gave no differences in comparison with the reference source arrangement. Although these experiments are somewhat limited, the results indicate that the choice of source arrangement is of practical importance. However, when regarding the results in this paragraph and keeping the standard transmission loss measurement procedure in mind, no specific simple source arrangement can be recommended for general application.

FIGURE 7

5. ADDITION OF ABSORBENTS TO THE BOUNDARIES

5.1. BASIC DISCUSSION

The basic idea behind the reverberant measurement methods is that we are dealing with adequately diffuse reverberant sound fields and that the direct field contributions are negligible. Therefore, in reference [30], it is stated that the average sound absorption coefficient $\bar{\alpha}$ should not exceed 0.06 over the frequency range of interest and above the frequency $f_a = 2000/V^{1/3}$, where V is the room volume in m^3 . In the same reference it is also recommended that below f_a it is usually appropriate to add some absorption to the reverberant chamber. However, in order not to get any semi-reverberant fields, the complementary recommendation is made, that $\bar{\alpha}$ should not exceed 0.16 in this low frequency region. Such a treatment increases the bandwidth of the resonance curves of the normal modes and decreases the effect of source position on the measured sound power. For chamber 3 this means that the reverberation times should exceed 2.2 seconds above 400 Hz and that they should exceed 0.8 seconds below this frequency limit.

A further reason for investigating the effects of adding absorbents to reverberant chambers, is that such an investigation makes it possible to form an opinion about the degree of sound field diffusion in more typical field measurement situations.

We know that an increase in absorption increases the importance of the direct field of the loudspeaker. By using the statistical sound field model, the following ratio can be stated between the mean square pressure of the direct field and that of the reverberant field.

$$\Delta(\underline{r}) = Q(\underline{r}) V / (100\pi r^2 T_{60}) \quad (6)$$

where $\underline{r}$ is the microphone position in the room, r is the distance between source and position (m), Q is the source directivity factor, T_{60} is the reverberation time (s) and V is the room volume (m^3). In reference [30] this relation is used to derive a minimum permissible distance between source and measurement positions, so as to avoid bias errors caused by the uninteresting direct field. This minimum distance is $0.08(V/T_{60})^{1/2}$. The specific limit is based on the assumptions that the source is non-directional and placed on the floor (not close to an edge or in a corner) and that the critical ratio is 1. This situation is somewhat specific and we do not know the critical value of Δ for our diffusional or mean square pressure measurements. As the experiments in this section are performed with a sound source placed in a corner, it is logical to insert $Q = 8$ instead of $Q = 2$ in equation (6), that is, we obtain the following alternative expression for the minimum distance:

$$r_{\min} = 0.16(V/T_{60})^{1/2} \quad (7)$$

The higher the frequency is, the more diffuse will be the reverberant sound field. Thus, from this point of view, the existence of relatively high values of $\Delta(\underline{r})$ must be considerably more critical the higher the frequency is.

5.2. EXPERIMENTS

Some measurements with additional absorption have been performed in the empty reverberation chamber 3. Both a concentrated area of absorbents and a randomized localization of absorptive patches are studied. The approximate values of the reverberation times and the respective Δ -ratios are shown in table 2. The Δ -values are derived for a typical microphone position i.e., for the middle of the room ($r = 4.0$ m).

TABLE 2

The results of the diffusional measurements are shown in Figure 8. Throughout these measurements no microphone position has been chosen closer to the reference source than 2.8 m. The reference source is placed in a corner. For ordinary sound pressure level measurements and for the most critical case in table 2, the recommended minimum distance between microphone positions and the source is 1.6 m (see equation (7)).

FIGURE 8

As we can observe in Figure 8, except for the 125 Hz situation, the values of ϵ_d have increased considerably as a result of the absorption added. When regarding the fact that the number of directional wave components or modes that build up the 125 Hz reverberant sound field is small, and that the Δ -values in table 2 consequently indicate that the direct wave component cannot be very dominating, the results at 125 Hz are not very surprising. Another measurement at 125 Hz, with some more random absorbents added ($T_{60} \approx 1.2$ seconds), gave the same response as the experiment above.

To get a further penetration of what has happened at the other frequency bands, some frequency response measurements have also been made. This complementary investigation has been done for the 250 Hz 1/3-octave band only, because both experimental and theoretical investigations have shown that the frequency irregularity does not give any additional information about the state of diffusion at high frequencies [4]. Quoting reference [4], "there is still the possibility that the frequency irregularity has independent significance at frequencies lower than Schroeder's limiting frequency". The respective spatially averaged mean-square pressure frequency response curves are shown in Figure 9.

FIGURE 9

The results are quite logical in comparison with classical room acoustics. The response of the chamber has become smoother by adding randomly located absorbents, and a natural and specific suppression of several resonances has been obtained by adding the concentrated sample of an absorbent at one of the chamber walls.

At the same time as the values of ϵ_d have been derived, values of a quantity called ϵ_n have also been produced. This quantity, described in reference [7], is basically a typical spatial variance of the measured cross-correlation functions. However, as distinguished from the derivation of ϵ_d , this variance is estimated by using the spatial average cross-correlation function as the expected function (see reference [7] equation (2.1)). On a comparison between the ϵ_d -values obtained in the experiments above and the respective ϵ_n -values, no significant difference was found in their interrelations. This points to the conclusion that, regarding ϵ_d , no essential change has occurred by altering the frequency characteristics. Hypothetically

this observation can be explained by the fact that no drastic large scale changes have occurred in Figure 9 and by the natural fact that ϵ_d must be essentially less sensitive to small scale changes in the frequency characteristics than to large scale changes. This reasoning is also supported by the fact that the increase in ϵ_d has also occurred at 500 and 1250 Hz, where the frequency representation must be still more even than in the 250 Hz case. Thus the alteration of ϵ_d must be explained by the directional compositions of the sound fields.

In comparison with the classical normal-mode theory the frequency response curve of the "concentrated absorption" case also exposes something about the directional characteristics of the sound field. A natural and specific selection has been obtained. This is quite consistent with the fact that the ϵ_d -values are higher in this situation than in the empty room case. For the "random absorption" case, however, the response curve does not disclose anything about the directional characteristics. What a broad frequency peak corresponds to in directional terms is not known. A theoretical way of attacking this reverberant sound field situation is to use acoustic rays from the source (geometrical acoustics). From this way of visualizing the sound field it is clear that the sound field determined by the absorptive patches will be less diffuse in a typical position than the empty room sound field.

Finally, we also know (see table 2) that the two cases with additional absorption both have direct field components and that these components, according to the discussion in section 5.1 above, are more important and devastating the higher the frequency is. As an elucidative example some sound pressure level measurements for the "random absorption" case at 1250 Hz deserve to be mentioned. In consistency with the statistical sound field model it was observed that contributions to the total sound pressure level from the direct

sound field were obtained as far out as 3.0 m from the reference source (the middle of the room is situated 4.0 m away from the source). Another result that also can be predicted by the statistical model was that the total sound pressure level was about 3 dB higher than the typical reverberant sound field level at the distance $r_{\min} = 1.6 \text{ m}$ ($\Delta = 1$).

Conclusively it can be stated that the ϵ_d -measurements agree qualitatively well with the classical theories, and that the addition of absorbents is devastating for the sound field diffusion, especially at high frequencies. The existence of absorbents suppresses important directional components of the reverberant sound field and amplifies the importance of the direct sound field in comparison with the reverberant wave components. No alteration of the sound field diffusion has been observed at the lowest frequency band, and when the "random absorption" experiment was repeated in the smaller reverberation chamber 2 (104 m^3), the same results as above were obtained (the results of this experiment are shown in Figure 12). When regarding the recommendation in reference [30], i.e., about how to add absorbents below a certain frequency f_a when performing sound power measurements, the results above point out that the frequency limit f_a seems to be too high, from the diffusional point of view ($f_a = 426$ and 393 Hz for chamber 2 and 3 respectively).

Finally the practical importance of the observed diffusional alterations is checked by the sound pressure level difference procedure. When measuring D with the triple window mounted in the transmission loss laboratory, the results shown in Figure 10 were obtained. As can be seen, significant alterations have been obtained at 125 and 1250 Hz, when the concentrated absorption sample is added to the reverberant chamber. Another D -experiment for the "concentrated

absorption" case and another wall construction gave (in comparison with the empty chamber case) the result that no differences were observed at 125 and 250 Hz, and that significantly higher values were obtained at 500 and 1250 Hz.

In accordance with the results reported in literature, it is thus found that the existence of a concentrated sample of an absorbent is damaging to the sound field diffusion. This holds for chamber 3 above 125 Hz. The diffusional experiments gave no indications about what to recommend for the 125 Hz situation.

Concerning the "random absorption" case no significant alterations have been observed in the D-experiments. But as it has been necessary, for practical reasons, to restrict the number of D-measurements, and as the measure of diffusion has shown to be considerably affected above 125 Hz, it is also recommendable to avoid "random absorption", at least at high frequencies.

FIGURE 10

6. STATIONARY REFLECTIVE PANELS

The line of thought behind treating the reverberation chamber with randomly hanging reflective panels is to make the simple boundary geometry more random and so reduce possible sound field degeneracy at low frequencies. But, as maintained in the literature survey, the experimental experiences indicate that the usefulness of stationary diffusers in ordinary hard-walled chambers is questionable for stationary sound fields. Only limited diffusional effects (at high frequencies) have been observed [17]. This circumstance is not unlogical when elucidated by the facts that the number of eigenfrequencies is not essentially altered, unless the chamber volume is changed, and that ordinary chambers do not have such simple shapes as discussed in connection with degenerate sound fields. This reasoning is fully supported by the results of the hard-walled chamber experiments shown in Figures 11 and 12. In Figure 11 no effects can be observed on ϵ_d (six reflective panels have been applied to the reverberation chamber 3). In Figure 12 a significant positive effect can be observed at 1250 Hz (the measurements were performed in the reverberation chamber 2 and an essentially larger relative panel area than in Figure 11 was used).

FIGURE 11

In literature and in section 5.2 we experienced that the addition of absorption to the empty chamber was devastating for the state of sound field diffusion. In the literature survey it was also maintained that stationary panels were significantly more useful the more absorption that was introduced into the chamber. Therefore, to penetrate this statement, the experiment in chamber 2 was completed

with a "random absorption" arrangement. The respective ϵ_d -results are shown in Figure 12. According to section 5.2 higher ϵ_d -values are obtained for the "random absorption" arrangement above 125 Hz, in comparison with the empty room case. We observe that the lower diffusivity for the "random absorption" situation can be improved at high frequencies by the use of randomly hanging panels. However, the resulting values are still considerably higher than the respective hard-walled chamber values.

FIGURE 12

As it appears from the presentation above, the use of stationary hanging panels has not produced any alteration of the undesirable low diffusivity at low frequencies. At high frequencies some effects have been experienced. This might be explained by the fact that the direct wave component has become scattered, that is, less dominating. Although only marginal effects are observed, it is interesting to check the practical importance of hanging panels, as they are commonly used in e.g., standard transmission loss measurements. Accordingly, some sound pressure level difference measurements are made, and the associated results are presented in Figure 13.

FIGURE 13

The case including both hanging panels and random absorption and representing significantly higher ϵ_d -values, also gives significantly different D-values at 500 and 1250 Hz compared with the empty chamber case. This observation gives further support to the conclusion that absorbents should be avoided at high frequencies.

In Figure 13 there is also a significant difference between the

empty chamber result and the result obtained when the hard-walled chamber was equipped with the stationary diffusers. This difference occurs at 1250 Hz. At the corresponding ϵ_d -experiment no difference was obtained (see Figure 11). These results are not unlogical, as the existence of the reflective panels must alter the sound field composition, although not necessarily the degree of sound field diffusion.

Summarily, the investigation above indicates that it is equally correct, from a diffusional point of view, to measure with as well as without stationary diffusers in reverberant chambers. However, various standard measurement results may be obtained, as the respective sound fields are differently composed. When measuring in a chamber equipped with absorbents, the results indicate that the use of hanging panels is preferable.

7. MOVEABLE REFLECTIVE PANELS

7.1. BASIC DISCUSSION

For the various situations studied above no drastic improvement of the measured sound field diffusion has been observed. This fact coincides with the general opinion about these devices (see section 2 above). Therefore we fix our last hope on the use of rotating diffusers and moving boundaries. Fortunately, the theoretical and experimental investigations concerned with such devices have shown significant positive effects. E.g., in a contributed paper at the 90th meeting of A.S.A., Tichy reported that it was found, with the aid of a theoretical model, that the spatial variance of the mean square pressure decreased by a factor of 0.25 - 0.8 when a time varying boundary was introduced. The summary of this paper, see reference [31], is the only report found in the literature about moving boundaries.

When introducing moveable elements into a chamber we can alter, either discretely or continuously, the boundary system of the chamber. Accordingly, it is possible to alter the eigenfrequencies and incidence angles of the dominating waves. By moving sufficiently large reflective elements within the hard-walled chamber, but keeping the excitational system unchanged, different reverberant sound fields will be obtained. Each of these sound fields quite naturally will have the same typical characteristics as have been observed for the stationary sound fields studied in the preceding sections. But now it is possible to perform an averaging of the measured values for the different sound fields (a so called room averaging). This means that we produce a mathematical superposition of the alternative

sound fields, and that the resonant character (see e.g., figure 4) of the sound fields will be averaged out. This result presupposes that the individual sound fields are significantly different from each other. Consequently, if the room averaging can be performed accurately, then the diffusion of the room averaged sound field will be essentially better than those of the individual sound fields.

Principally, the direct way to achieve the alterations of the sound fields is to move discretely some sufficiently large reflective elements within the chamber. The elements can also be moved continuously. However, in order not to introduce any new excitational frequency components, this continuous movement must be so slow that the succession of sound fields produced are the same as those steady state sound fields that would have been obtained if the elements had been successively and discretely moved along the same path. Such an averaging involves a lot of redundant information, because a small displacement of a boundary quite naturally does not produce a very different sound field. This framing of the problem will be partly studied in the sections 7.2 and 7.3 below.

When regarding a discrete measurement position this room averaging is quite naturally time consuming. E.g., if we choose to move a reflective element into 5 discrete positions, then the integration times will increase by a factor 5. On the contrary, the room averaging will decrease the spatial variance and the need of taking many spatial samples. So, this way of producing a more diffuse measurement situation (a more uniform representation of incidence angles and frequencies) must not necessarily be very time consuming.

7.2. MODEL EXPERIMENTS

No detailed recommendation is presented in literature about how to construct a moving diffuser. Only some general recommendations are given. E.g., it is often stated that a moving diffuser must have dimensions of the same magnitude as the wavelength. Therefore some model experiments were performed before the full-scale diffuser system presented in section 7.3 was realized.

A 1:10 scale model of the reverberation chamber 1 was used for the experiments. In order to get results comparable with other authors', it was decided to study the normalized mean square pressure spatial variance and to use both 1250 Hz pure tone and 1/3-octave band noise excitation. The 1250 Hz signals were chosen as the main interest has been devoted to low frequency sound fields.

When starting the first measurements, the pure tone experiments, the observation was made that it was difficult to maintain a stationary sound field. With several laborious trial and error experiments it could be concluded that it was necessary to use a stable pure tone generator (ANADEx type: Frequency Synthesizer FS-600R), that the temperature conditions had to be relatively stable and that it was important not to open the model chamber during the spatial sampling. To avoid the opening of the chamber when performing the spatial sampling with the measurement microphone, a special adjustable rod was mounted in the transparent front wall of the chamber. A time variance was continuously derived with the aid of a fixed reference microphone, so as to check that the system was stable. When regarding the principles above, normalized mean square pressure time variances of 0.001 - 0.009 were obtained. In comparison with the spatial variances obtained below the values are sufficiently small,

that is, the test values can be regarded as describing the spatial characteristics only. When instead the chamber was opened every time before a new spatial sample was taken, a time variance value of e.g., 0.22 was obtained. I.e., a value of the same magnitude as the spatial variances was observed. This example clearly reveals the importance of using a stable chamber arrangement.

The reason why the system is that sensitive to small changes in e.g., the excitation frequency, is that the frequency response of the chamber, as illustrated below, is very irregular. The reverberation time in the model chamber is about 1/10 of that in the full-scale chamber, and the bandwidth of a typical room resonance is approximately $2.2/T_{60}$. I.e., the frequency response of the model chamber must resemble those shown in Figure 4. When using Figure 4 for this discussion we only need to multiply the values of the frequency axis by the factor 10.

When experimentally checking the loudspeaker (PHILIPS type: 57 mm diameter, high quality tweeter AD 2071/T4) it could neither be classified as an extremely high nor an extremely low impedance type of loudspeaker.

The final model reverberation chamber arrangement is illustrated in Plate 1.

PLATE 1

To estimate experimentally the spatial variances, the following expression was used:

$$V_s^2 = \frac{\sum_{i=1}^n \left(\hat{\psi}_i^2 - \frac{\sum_{i=1}^n \hat{\psi}_i^2}{n} \right)^2}{\left[\left(\frac{\sum_{i=1}^n \hat{\psi}_i^2}{n} \right)^2 (n - 1) \right]} \quad (8)$$

where V_s^2 is the normalized mean square pressure spatial variance, $\hat{\psi}_i^2$ is the mean square pressure estimate, a subscript i indicates that the quantity concerned is that at the i th randomly chosen discrete microphone position and n is the number of spatial samples. Every spatial sample has been obtained by performing a long time averaging of the sound pressure squared. As indicated above, and when continuously moving reflectors are used, this averaging time should be chosen to enclose the interval of room averaging. Consequently, when the pure tone rotating diffuser experiments presented in Table 3 were performed, the averaging time was equal to the time of one full revolution, i.e., 95 seconds.

TABLE 3

As can be seen from Table 3 the best rotating diffuser configurations imply that the spatial variance has decreased by a factor about 0.3. The respective factor obtained from some considerably less extensive 1/3-octave band noise experiments was 0.7. In accordance with previous reports in literature, the results have been better, the more the rotating diffuser takes up of the room. However, it is unpractical to use voluminous rotating reflectors. From this point of view a moving boundary system (moving a ceiling and/or one or more walls) seems much more attractive, because within such a chamber there can always be reserved a large free inner reverberant space. Besides the associated practical advantages, this circumstance also makes it possible to perform a sufficient spatial sampling and to perform other investigations that demand a free space.

A moveable boundary system is simple to realize in a model chamber. Therefore such a system has also been investigated. A continuously

moveable suspended ceiling (taking up 93 % of the ceiling area) and two additional inner discretely moveable walls (taking up 72 % and 52 % of the respective wall areas) composed the boundary system. The total one-sided area of the reflectors was $9.5 \lambda^2$, where λ is the wavelength at 1250 Hz and 20° C.

To check the efficiency of using a continuously moving ceiling, the experiments presented in Figure 14 were performed. Using a maximum displacement of $\lambda/2$ in the measurements, it was also possible to derive results representable for other displacements. These results are also presented in Figure 14. V_s^2 has decreased by a factor 0.19 and 0.50 for the pure tone and 1/3-octave band situations respectively. Concerning the pure tone response of the chamber it seems to be effective to use a maximum displacement of as much as $\lambda/2$. However, the irregular pure tone curve obtained indicates the natural fact that the successively generated sound fields are not independent and that much irrelevant and possibly damaging information is obtained when such a continuous room averaging is used.

FIGURE 14

The experiments above were completed by discretely moving the wall reflectors. A room averaging was investigated, which included five discrete positions of the additional walls, besides the continuously moving ceiling (maximum displacement $\lambda/2$). I.e., the averaging time was increased by a factor 5. In comparison with the empty chamber case the result was that V_s^2 decreased by a factor 0.10 and 0.27 for the pure tone and 1/3-octave band situations respectively.

With the results above and the practical advantages in mind, it was decided to build a full-scale moveable boundary system like

that investigated above. Therefore, to get an indication about how much the diffuser surfaces must weigh, some final model experiments were performed with the ceiling. The results of these measurements are presented in Figure 15. A surface mass of 0.5 kg/m^2 seems to be quite sufficient for the very situation studied. Using the mass law to obtain a recommendation for 125 Hz, this result implies that the full-scale panels should weigh at least 5 kg/m^2 . This observation coincides with the recommendation given by Schultz in reference [4], see section 2 above.

FIGURE 15

7.3. A FULL-SCALE MOVEABLE BOUNDARY SYSTEM

To affect all room modes in a rectangular reverberation chamber, it is sufficient to alter the location of three mutually perpendicular boundaries. A moveable boundary system has been constructed within chamber 3, making a so indicated room averaging possible. The chosen reflective panel arrangement is presented in Figure 16.

FIGURE 16

A 0.7 mm steel-plate with a 1.3 mm free viscoelastic layer (Dempson damping panel type P) was chosen for the construction of the reflective boundaries. The surface mass of this composite panel is 6.7 kg/m^2 , that is, in comparison with the discussion in section 7.2, sufficiently large. The geometrical characteristics of the additional boundaries are summarized in table 4. Owing to the extension of the test opening of the transmission loss laboratory it has been necessary to choose relatively small maximum displacements for the boundaries Nos. 2 and 3.

TABLE 4

Quite naturally it is necessary, when constructing a continuously moving boundary system, to raise and fulfill demands upon variableness of displacement velocity and maximum background noise level. Because of this it is easier and less expensive to build an accurately functioning system based on discrete displacements. When using a discretely moveable boundary system, the room averaging is obtained with the aid of a set of discrete stationary reflector locations.

That is, the boundary reflectors are not displaced when the mean square pressure measurements are performed. Because of the practical reasons outlined above, the system was constructed only for discrete movements.

A partition wall made of a 13 mm thick gypsum board and mounted in the test opening, flush with the frame wall of the source room, was used for all the experiments in this section. As sound source the reference loudspeaker was used.

When performing a room averaging as indicated above, the discrete mean square pressure spatial samples are to be obtained according to the expression

$$\hat{\psi}_i^2 = \sum_{j=1}^m \hat{\psi}_{ij}^2 / m \quad (9)$$

where m is the number of boundary configurations used and the subscripts i and j indicate that the quantity concerned is that of the i th microphone position and the j th boundary configuration.

When investigating the effectiveness of such a room averaging it is interesting to study the spatial variance of $\hat{\psi}_i^2$:

$$\text{Var}[\hat{\psi}_i^2] = E\left[\left(\hat{\psi}_i^2 - E[\hat{\psi}_i^2]\right)^2\right] \quad (10)$$

where $E[\]$ denotes the expected value of $[\]$. By use of equation (9), equation (10) can be rewritten as

$$\text{Var}[\hat{\psi}_i^2] = \left(\sum_{j=1}^m \text{Var}[\hat{\psi}_{ij}^2] + 2 \sum_{j=1}^{m-1} \sum_{k=j+1}^m \text{Cov}[\hat{\psi}_{ij}^2, \hat{\psi}_{ik}^2] \right) / m^2 \quad (11)$$

where the covariance $\text{Cov}[\hat{\psi}_{ij}^2, \hat{\psi}_{ik}^2] = E\left[\left(\hat{\psi}_{ij}^2 - E[\hat{\psi}_{ij}^2]\right)\left(\hat{\psi}_{ik}^2 - E[\hat{\psi}_{ik}^2]\right)\right]$.

As can be seen in equation (11) the spatial variance resulting from the room averaging is composed of two terms. The first term represents the average spatial variance of the individual sound fields divided by m . The second term represents the sum of the mutual covariances divided by m^2 . If the sum of covariances is zero, the following familiar expression will be obtained:

$$\text{Var}[\hat{\psi}_i^2] = \sum_{j=1}^m \text{Var}[\hat{\psi}_{ij}^2]/m^2 \quad (12)$$

The lower the resulting spatial variance is, the better measurement precision will be obtained for a given amount of sampling. Hypothetically, this will also affect the diffusion of the room averaged sound field analogously. Consequently, equation (11) indicates one way to search for a suitable set of reflector configurations. The various reflector locations shall be chosen so that the associated sum of covariances is minimized. The various terms in equation (11) can be measurementally estimated according to

$$\text{Var}[\hat{\psi}_{ij}^2] = \frac{n}{\sum_{i=1}^n} \left(\hat{\psi}_{ij}^2 - \frac{n}{\sum_{i=1}^n} \hat{\psi}_{ij}^2/n \right)^2 / (n - 1) \quad (13)$$

$$\text{Cov}[\hat{\psi}_{ij}^2, \hat{\psi}_{ik}^2] = \frac{n}{\sum_{i=1}^n} \left(\hat{\psi}_{ij}^2 - \frac{n}{\sum_{i=1}^n} \hat{\psi}_{ij}^2/n \right) \left(\hat{\psi}_{ik}^2 - \frac{n}{\sum_{i=1}^n} \hat{\psi}_{ik}^2/n \right) / (n - 1) \quad (14)$$

In order to find a set of reflector configurations that effectively gives a small resulting variance, a series of measurements were performed, in which stable 125 Hz 1/3-octave band noise excitation was used. 30 different boundary configurations were successively

investigated and for each of the associated sound fields the spatial mean square pressure values were measured by using a fixed array of 25 randomly chosen microphone positions. From the experimental values obtained, the $435 \left(= \sum_{j=2}^{30} (j-1) \right)$ unique covariances were estimated according to equation (14). From the information thus obtained, it was possible to select a set of five boundary configurations, which gave a normalized spatial variance value of 0.026. When comparing this value with the typical normalized spatial variance values obtained for the five individual sound fields, it was found that the room averaging had decreased the spatial variance by a factor of 0.14-0.44. Furthermore, no essential covariance contribution was obtained, that is, the room averaging produced was effective for this specific situation. The discussed normalization is performed by the respective squared spatially averaged mean square pressure values.

The effectiveness of using this specific set of boundary configurations has also been investigated for the 250, 500 and 1250 Hz 1/3-octave bands. For these experiments the room averaging resulted in a decrease of the spatial variances by a factor of 0.22-0.61. These results logically are not quite as positive as for the 125 Hz situation. When investigating the associated effects in the receiving room of the transmission loss laboratory, the interesting thing was found, that the spatial variances became significantly lower in this chamber too. The spatial variances decreased by a factor of 0.11-0.95. Although the boundary locations of chamber 3 have been altered considerably, the typical spatial variances of the individual sound fields were in good agreement with the respective values for the empty reverberation chamber.

After that the effects of the room averaging procedure on the diffusivity were investigated. The associated results are presented in Figure 17. To get a basis for the evaluation of the improvements

obtained, the diffusivity values for each one of the five individual sound fields have also been derived. When comparing these values with the reference empty chamber values in Figure 6, it can be seen that the diffusivity values have increased significantly at 500 and 1250 Hz. This is not very surprising, when regarding the fact that the establishment of the moveable boundary system have decreased the reverberation times to about 2.8 seconds. Compare with the discussion of the absorption experiments presented in Figure 8.

FIGURE 17

As can be seen from Figure 17, when superposing the five sound fields and applying the room averaging procedure, the diffusivity values became much lower. Referring to the average values for the five individual sound fields, the ϵ_d -values decreased by a factor of 0.45-0.65. In comparison with the outcome of the preceding experiments in chamber 3 (see e.g., Figure 6), the resulting values are extraordinarily low at 125 and 250 Hz. Then, when taking into consideration the unfortunate circumstance that it was necessary to restrict the maximum displacements for reflector No. 2 and No. 3 (see table 4), the results obtained above must be regarded as full of promises at low frequencies.

Finally the practical importance of using different boundary configurations within the source room is checked by the sound pressure level difference procedure. When measuring D with a masonry board mounted as partition wall, the results shown in Figure 18 were obtained. Significant differences can be seen at 125 Hz. From a statistical and logical point of view, none of the individual boundary configurations produces a sound field that is much superior to the others. The output

of a D-measurement for an individual boundary configuration depends on the specific irregular angular and frequency distributions of the impinging sound field, in combination with the respective transmission characteristics of the specific partition wall studied. Therefore this experiment implies that the most accurate and representative value of D will be obtained with the room averaging procedure.

Although this experiment is limited to one specific wall, it might be important to emphasize that these results also give an idea about what can be expected for the low frequency region, when measuring in laboratories with source rooms that are differently shaped.

FIGURE 18

8. CONCLUSIONS

Concerning the diffusional experiments, the results have shown to be qualitatively consistent with existing theories. This circumstance confirms that the specific diffusivity measure ϵ_d has been satisfactorily designed. Furthermore the results have also been in good agreement with what is stated in literature about these matters. Summarily, the use of absorbents has affected the diffusion negatively at high frequencies; stationary hanging panels have improved the diffusion of the empty hard-walled chamber only in the small chamber 2 and at 1250 Hz; some specific loudspeaker arrangements have given considerably better values, and the most drastic low frequency improvements have been obtained by performing a room averaging with the aid of a specially designed discretely moveable boundary system. Although the investigated moveable boundary system is quite different from a rotating diffuser, its functioning is based on the same idea. That is, by altering the locations of the three boundaries, the eigenfrequencies and the incidence angles are altered.

The main reason to the successfulness of the moveable boundary system is that it is possible to use comparatively large moveable reflectors without intruding too much on the inner reverberant space. This is the most noticeable advantage such a system has in comparison with an accurately working, i.e., large, rotating diffuser. To present some further information about the system, it should be mentioned that it was not necessary to use complicated mechanical constructions and that the costs of material reached the total amount of \$ 2.500 (including the electrical drifting but excluding any advanced position control system).

When regarding the outcome of the sound pressure level difference

measurements, it is necessary to conclude that significant and practically important effects have been observed for all the individual alterations investigated. This result emphasizes that these matters should be considered at the development of future standards. For most of the cases the diffusional experiments indicate which situations are preferable.

The main interest in this paper has been devoted to the sound fields and their characteristics. Therefore the associated results are applicable also when discussing other standard measurement methods, e.g., the methods for determination of sound power levels of small sources in reverberation rooms. Also, as the experiments, with which the practical consequences of the sound field alterations were checked, had a limited extent, further research work is recommendable on this subject.

ACKNOWLEDGEMENTS

This paper is the last one in a series of three, that presents a research work dealing with the diffusion of reverberant sound fields (the first two parts of this work are presented in the references [7] and [25]). The research work has been financially supported by grant C865 from the National Swedish Council for Building Research.

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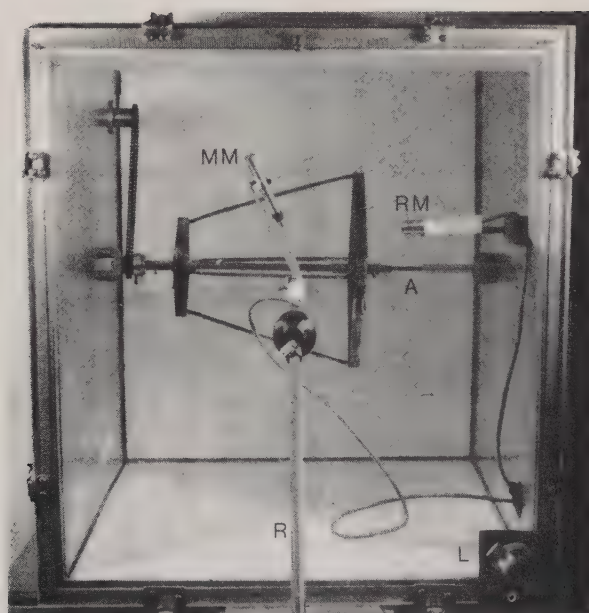


PLATE 1. A photograph of the model reverberation chamber arrangement. The microphone (MM) with which the spatial sampling is performed, is handled from outside with an adjustable rod (R) mounted in the plexiglas front wall of the closed chamber. The reference microphone (RM) and the rotatable axis (A) are magnetically attached to the side walls (the axis is here equipped with a truncated and uncovered pyramid shell). The loudspeaker (L) is externally mounted in the lower right corner.

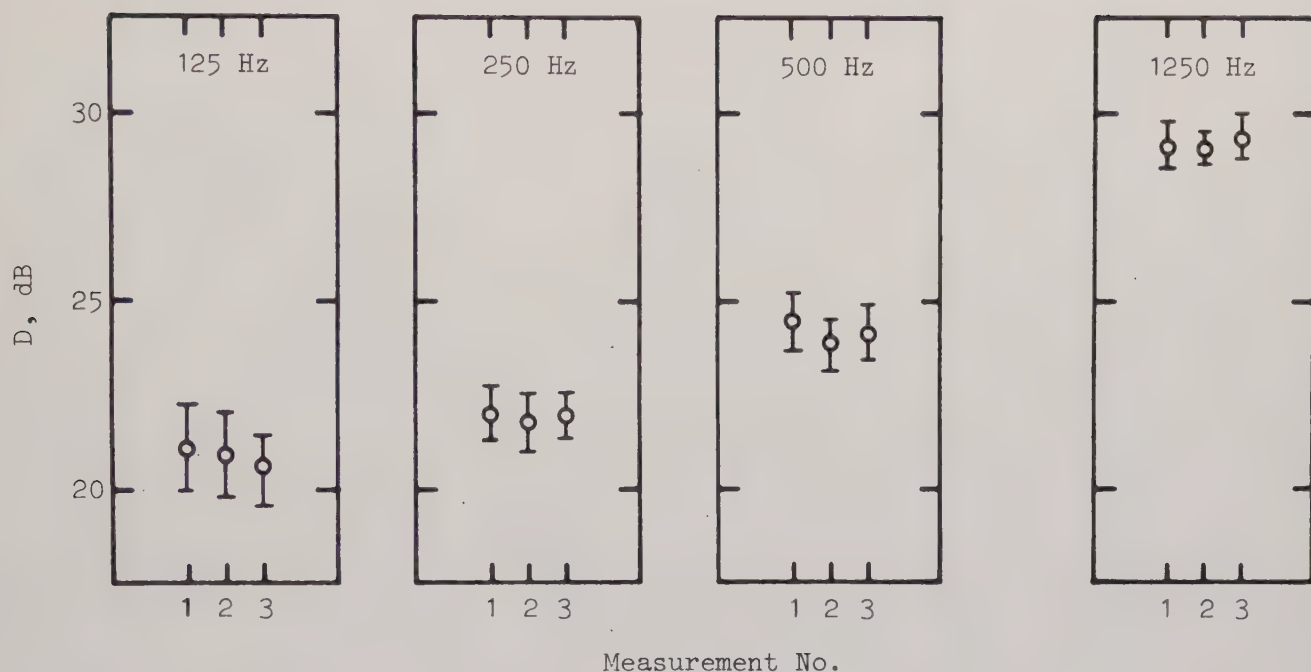


FIGURE 1. Three successively and independently obtained D-values (o), including estimated 95 % confidence intervals (—), at one and the same transmission loss laboratory condition for the 125, 250, 500 and 1250 Hz 1/3-octave bands.

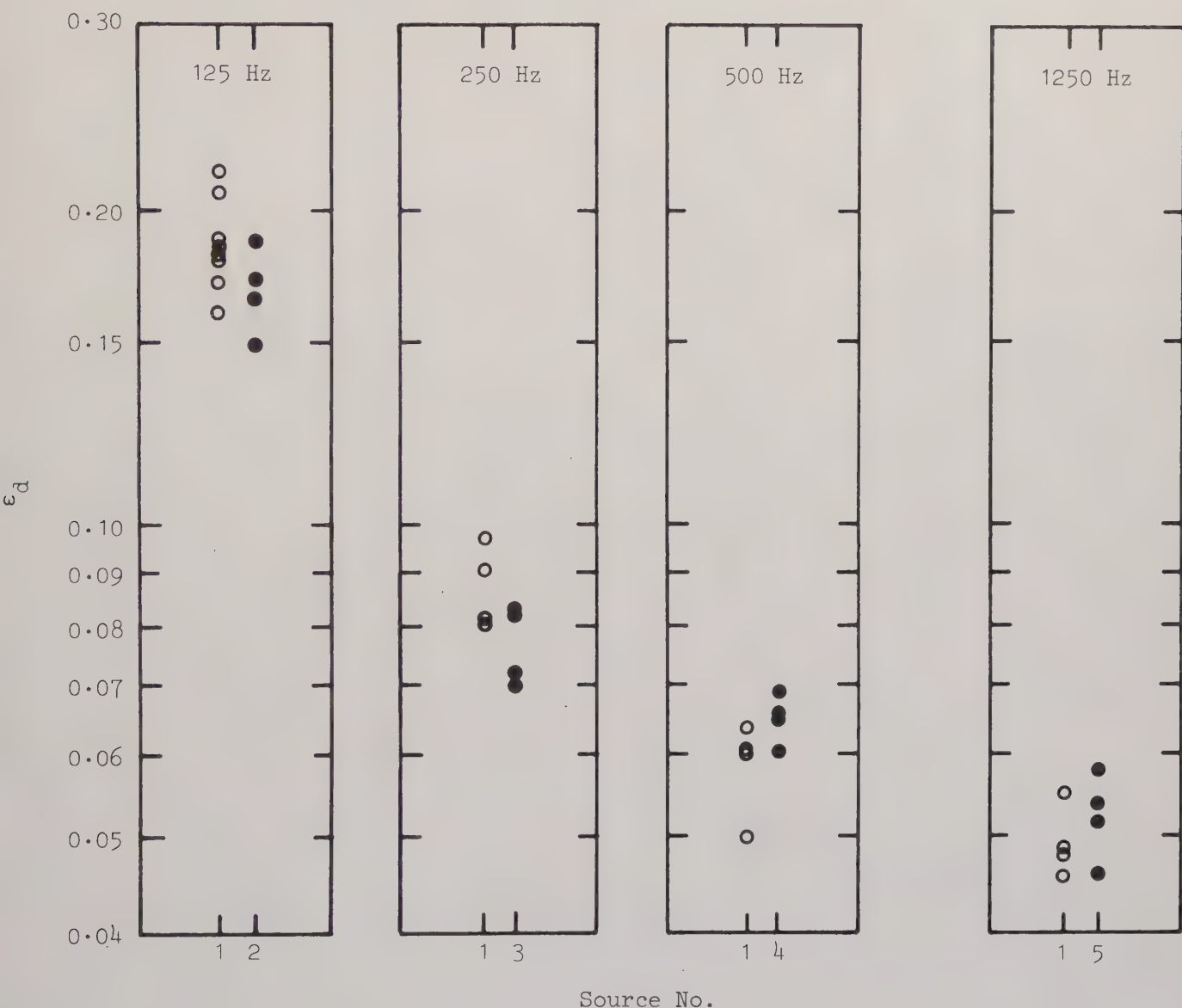


FIGURE 2. Measured diffusivity values for some different sound fields in the empty reverberation chamber 1. A comparison between results obtained with the reference source and some small loudspeakers. All the loudspeakers were placed as far into a corner as possible and at 45° to walls and floor.

Source numbers: 1, the reference source - a Goodman Axiom 301 mounted in a lined cabinet ($0.44 \times 0.44 \times 0.28 \text{ m}^3$); 2, Goodman Axiom 301 unit (diameter 0.31 m, open-back); 3, Goodman Maxim ($0.26 \times 0.18 \times 0.14 \text{ m}^3$); 4, Sinus FQ-3299 VQ unit (diameter 0.082 m, open-back); 5, Peerless MT 225 HFC Tweeter (diameter 0.056 m, closed back).

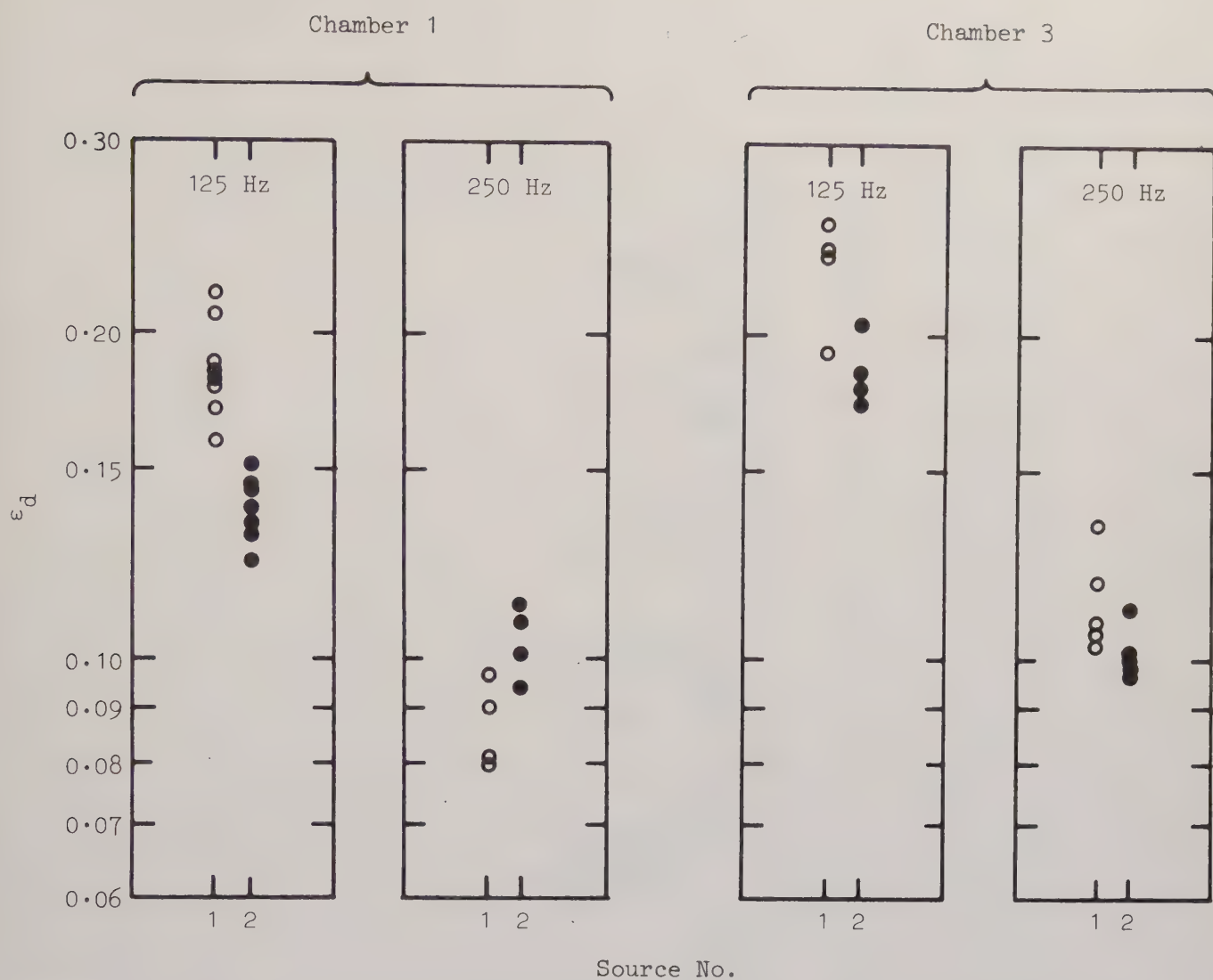


FIGURE 3. Measured diffusivity values for some different sound fields in the empty reverberation chambers 1 and 3. A comparison between results obtained with the reference source and a bass-loudspeaker.

Source numbers: 1, the reference source; 2, Goodman Sela Axiom 80 mounted in A.R.U. 172 enclosure (height 0.91 m, width 0.45 m, depth 0.25 m), horizontal loudspeaker axis 45° to walls.

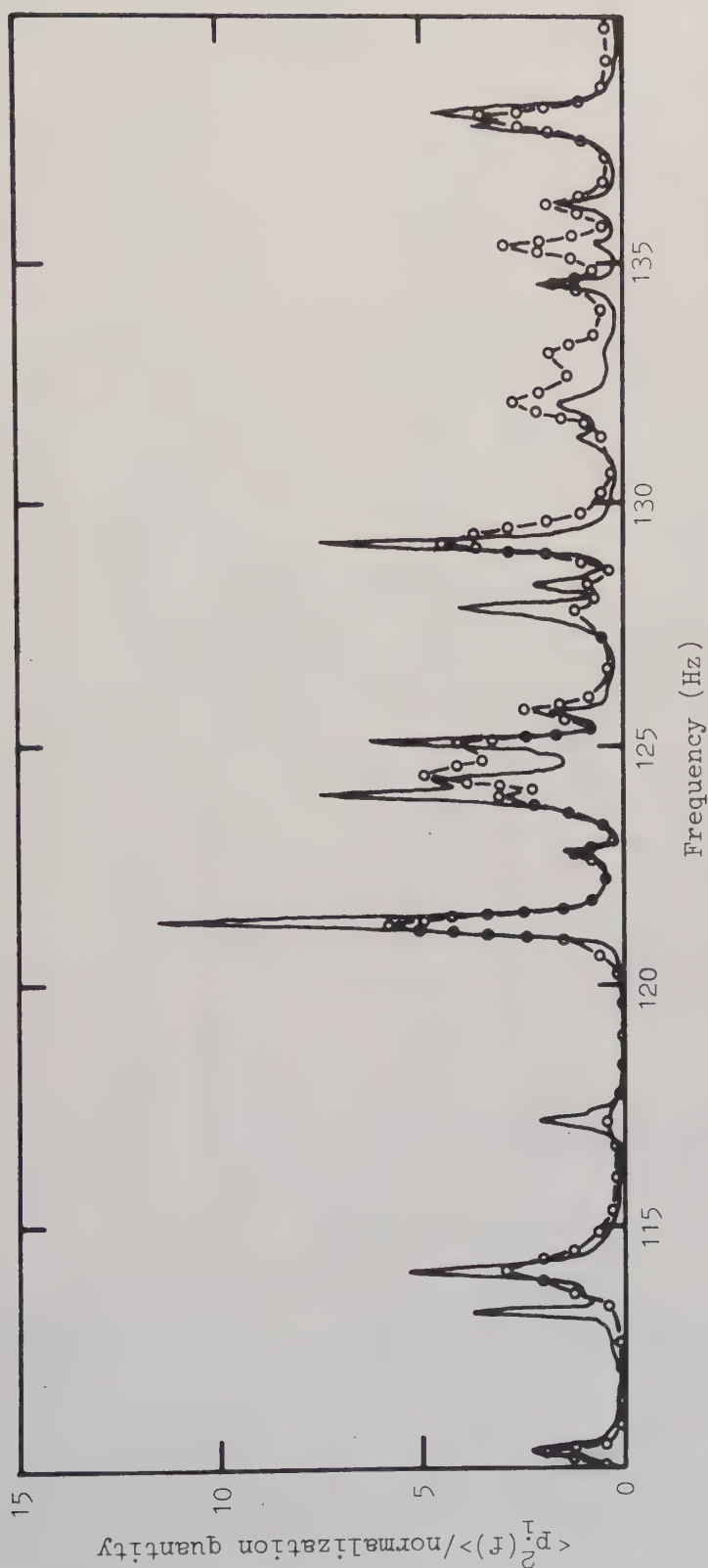


FIGURE 4. A comparison between two normalized mean-square pressure frequency response curves ($\langle p_1^2(f) \rangle / \text{normalization quantity}$) obtained for the empty reverberation chamber 1 excited with two different loudspeakers: —, the reference source; o-o-o, source No. 2 in Figure 3. The respective curves have been obtained by averaging the mean-square pressure response curves measured in 21 and 27 randomly chosen microphone locations.

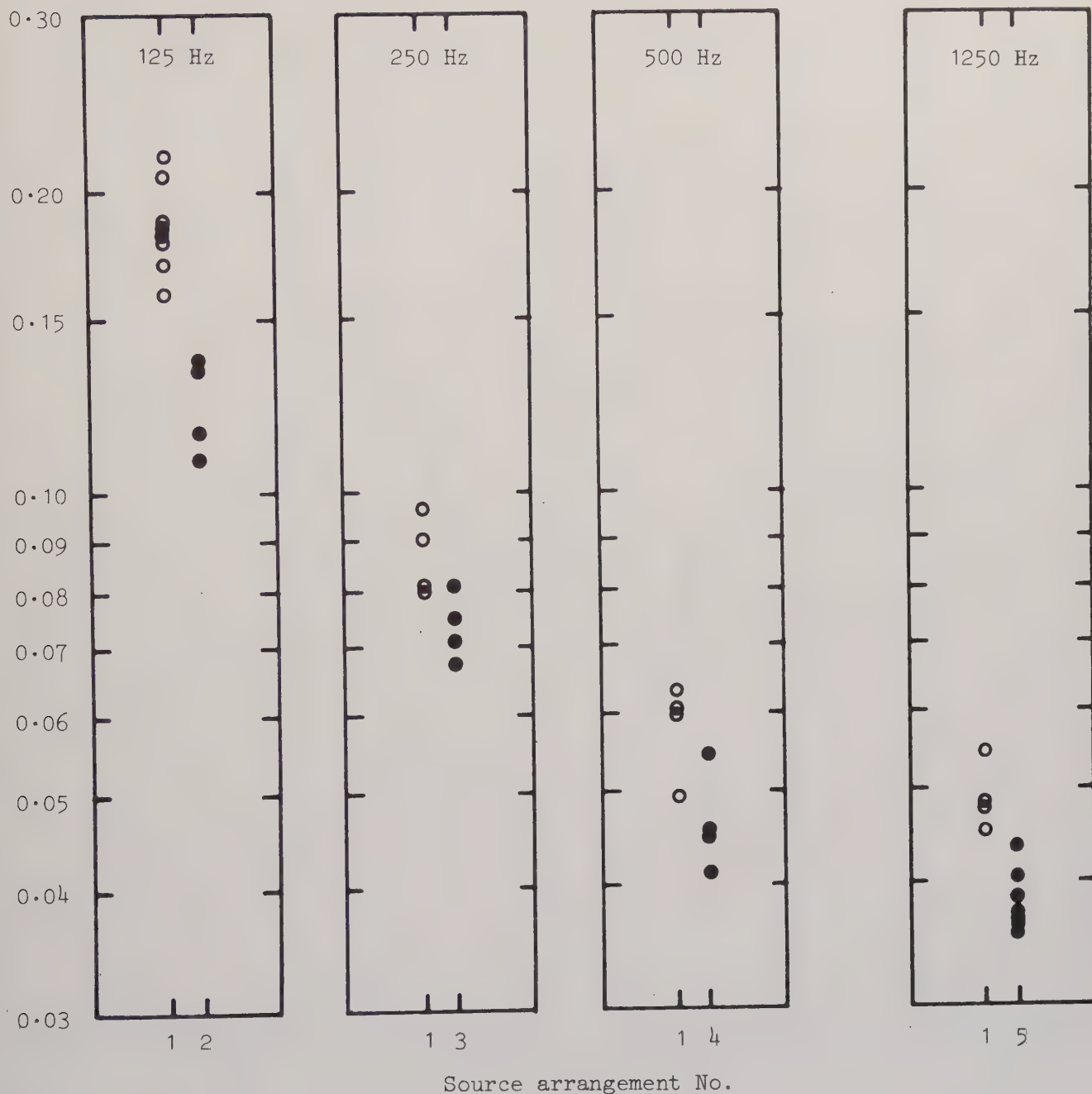


FIGURE 5. Measured diffusivity values for some different sound fields in the empty reverberation chamber 1. A comparison between results obtained with the reference source and some two-loudspeaker arrangements. When two speakers are used simultaneously, one of the speakers is placed in a corner (at 45° to walls and floor) and the other one is placed at a room edge (at 70° to wall and 45° to floor). Source arrangements: 1, the reference source; 2, Goodman Sela Axiom 80 mounted in A.R.U. 172 enclosure + Goodman Axiom 301 mounted in an open-back cabinet; 3, two Goodman Maximes; 4, Sinus FQ-3299 VQ unit + Sinus H-8099 Y5 unit (diameter 0.20, open-back); 5, two peerless MT 225 HFC Tweeters.

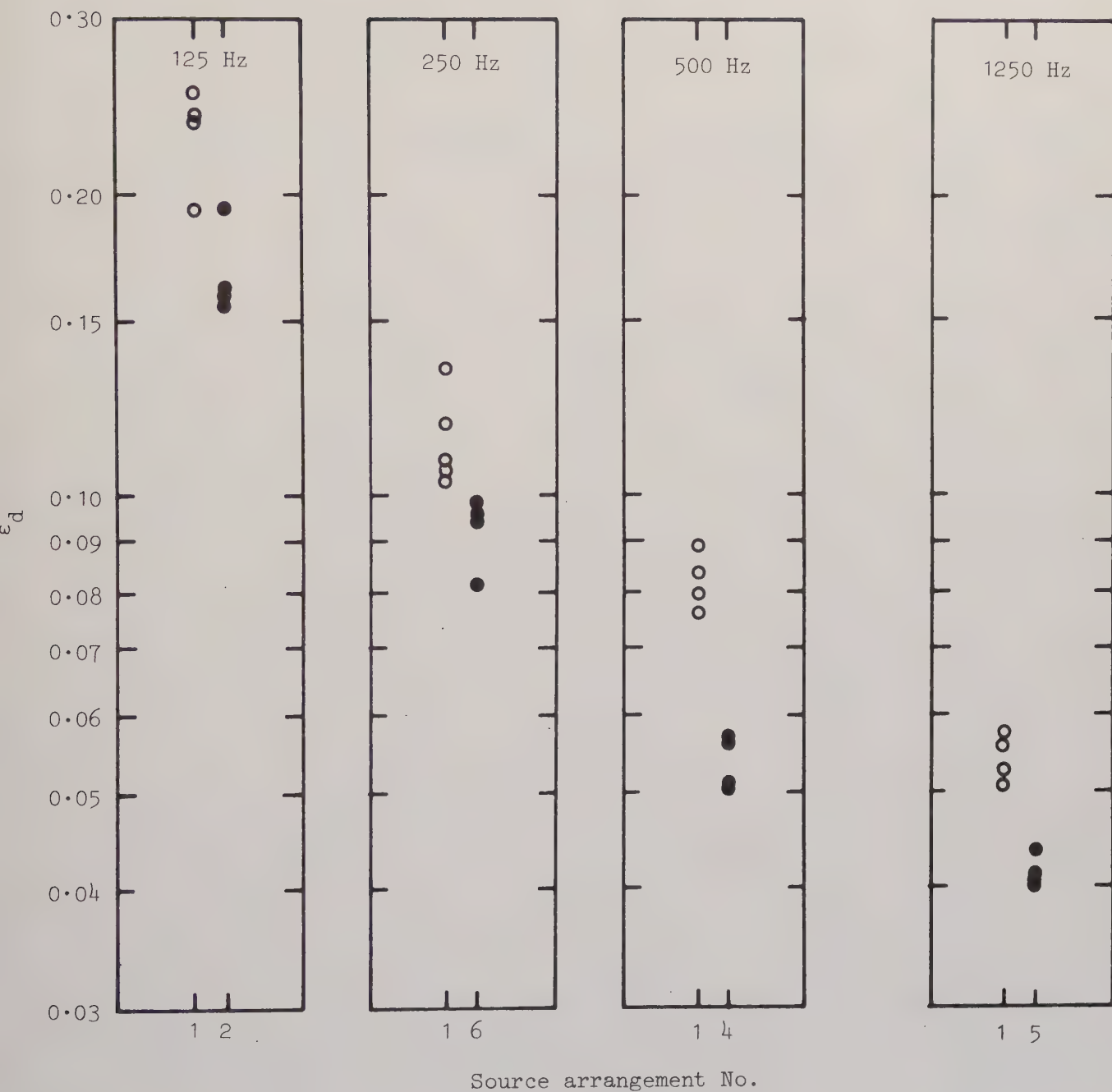


FIGURE 6. Measured diffusivity values for some different sound fields in the empty reverberation chamber 3. A repetition of the experiment presented in Figure 5 with the exception that the source arrangement denoted 3 has been exchanged for the source arrangement: 6, Goodman Axiom 301 mounted in an open-back cabinet + Sinus H-8099 Y5 unit.

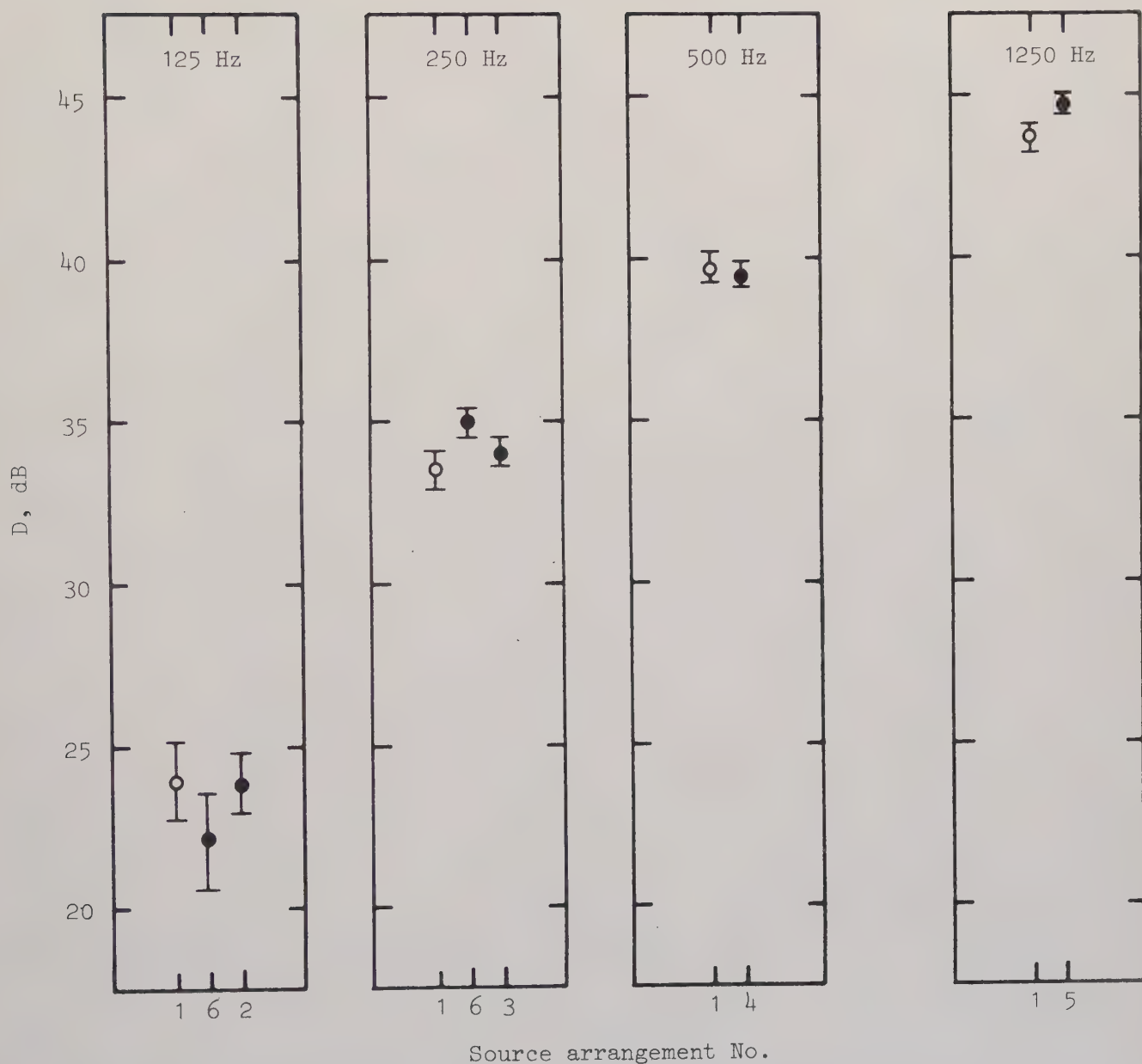


FIGURE 7. Comparison between sound pressure level difference values obtained in the transmission loss laboratory when measuring on a $1.2 \times 1.4 \text{ m}^2$ triple window (pane thicknesses = 6, 3 and 4 mm, airgaps = 20.5 and 0.5 mm) and when some different source arrangements are used in the empty reverberant source room (chamber 3). Source arrangement: 1, 2, 3, 4 and 5, see Figure 5; 6, the bass-loudspeaker (source No. 2 in Figure 3).

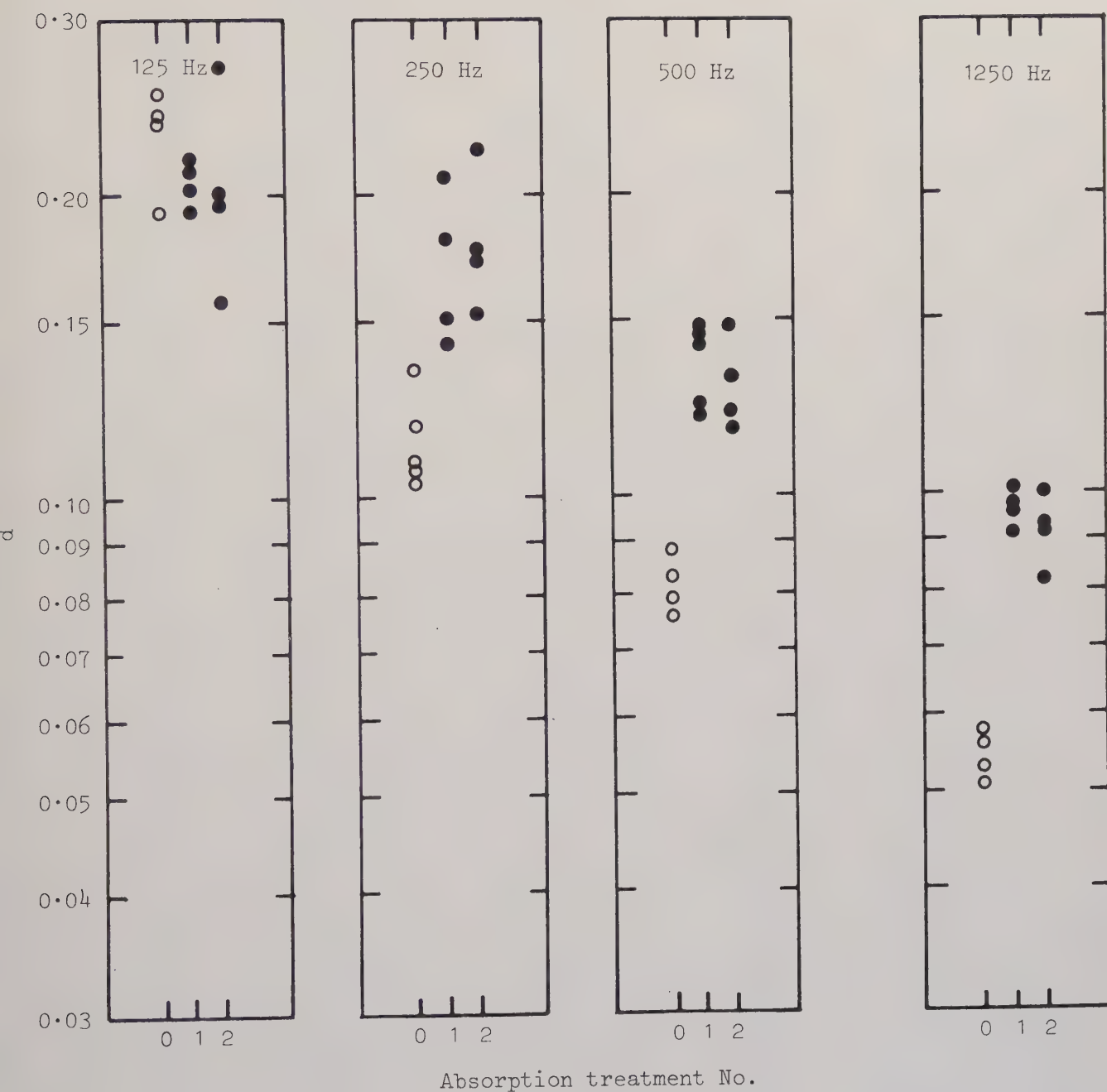


FIGURE 8. Comparison between measured diffusivity values for the empty reverberation chamber 3 and the same chamber equipped with additional absorption.

Absorption treatment: 0, the empty chamber reference case; 1, 10 absorptive patches randomly added to the boundaries (total area of porous absorbents $\approx 9 \text{ m}^2$); 2, 7.5 m^2 of a 0.12 m thick porous absorbent placed 0.26 m in front of one of the walls.

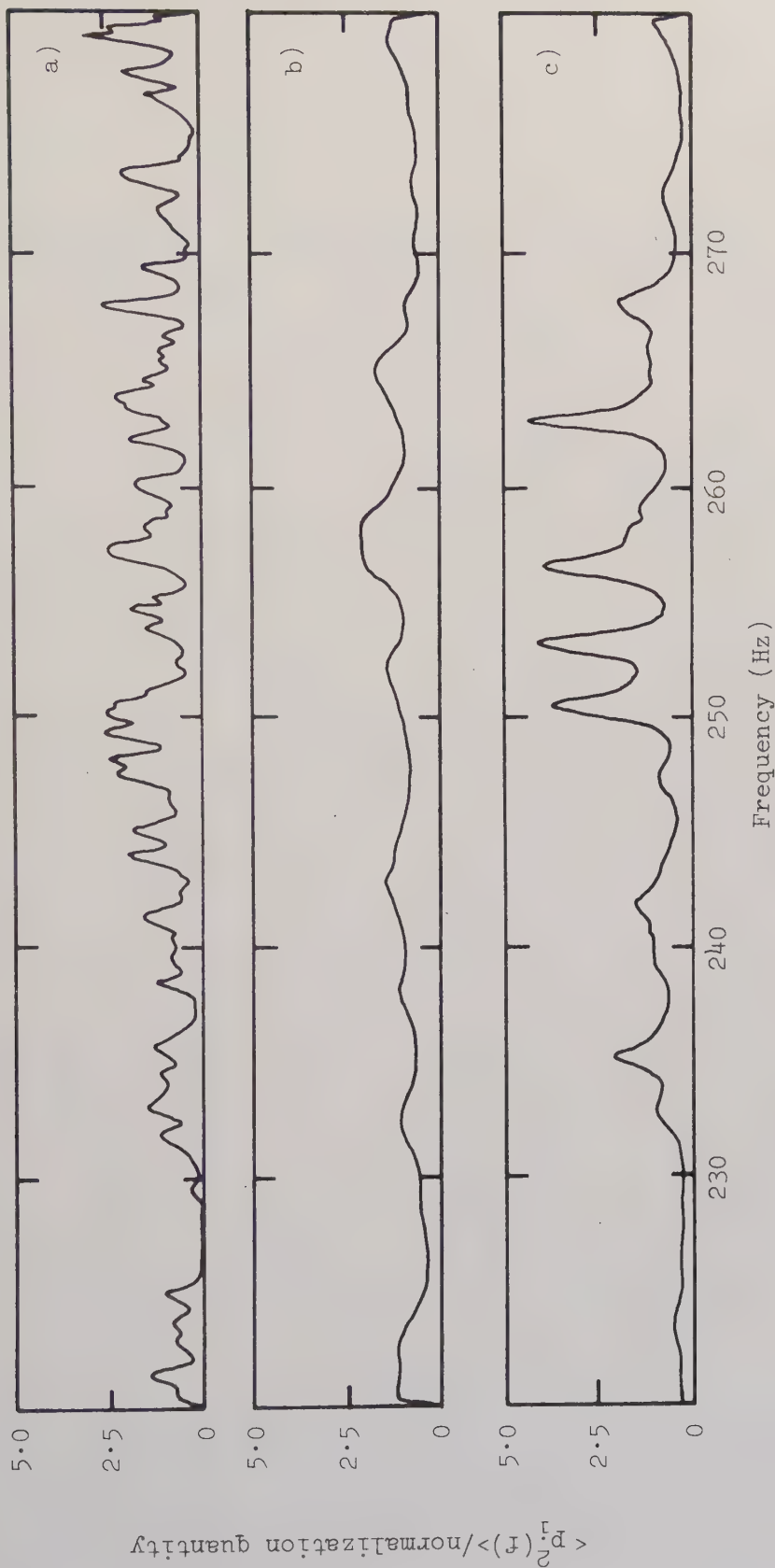


FIGURE 9. A comparison between three normalized mean-square pressure frequency response curves $\langle p_i^2(f) \rangle /$ normalization quantity) obtained in the reverberation chamber 3 at three different absorptive conditions. Diagram a represents the empty chamber reference case, diagram b the "random absorption" case and diagram c the "concentrated absorption" case. The respective curves have been obtained by averaging the mean-square pressure response curves measured in 24, 28 and 24 randomly chosen microphone locations.

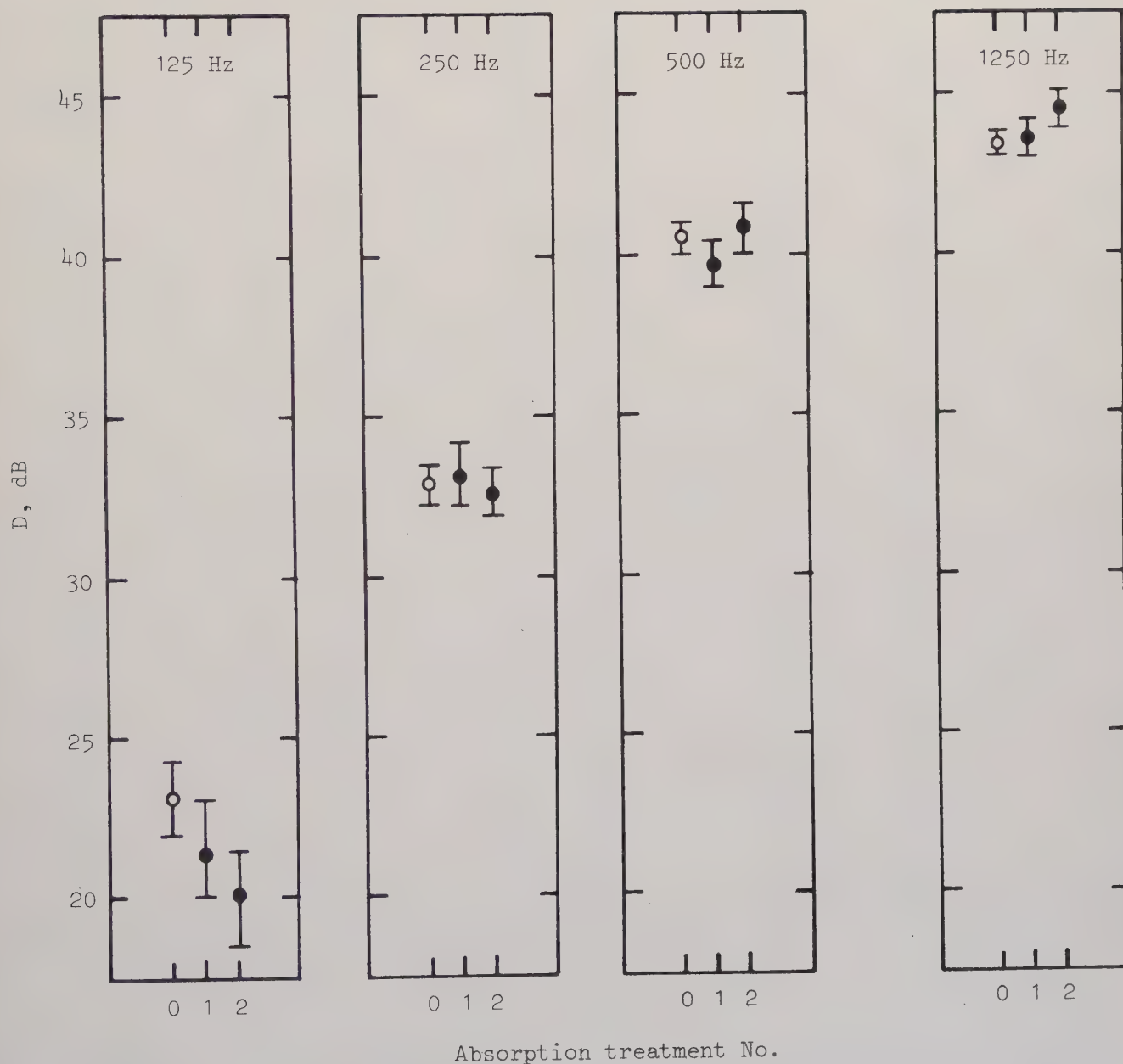


FIGURE 10. Comparison between sound pressure level difference values obtained in the transmission loss laboratory when measuring on the same window as in Figure 7 and when three different absorption treatments are applied to the empty reverberant source room (chamber 3). Absorption treatments: see Figure 8.

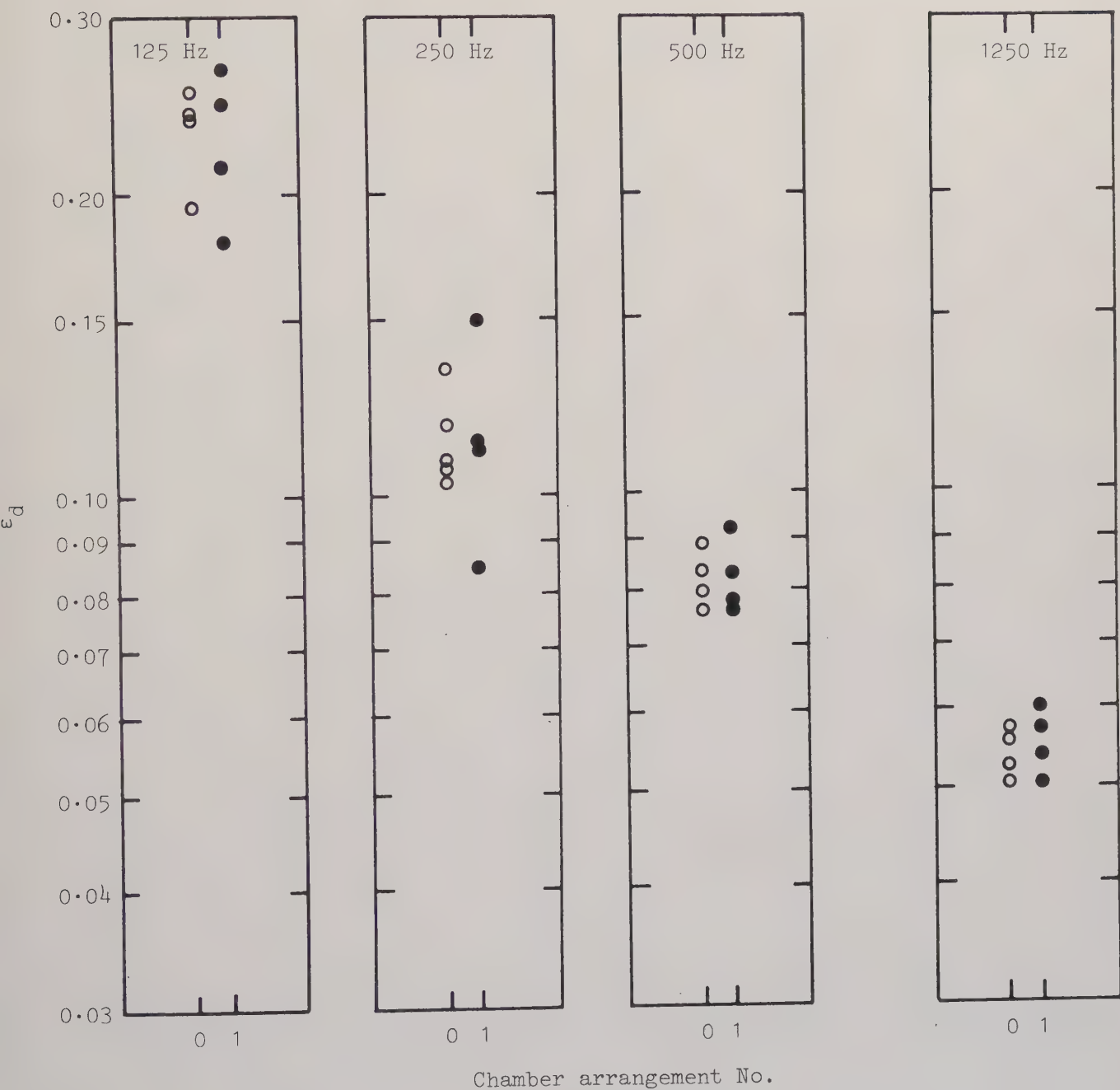


FIGURE 11. Comparison between measured diffusivity values for the empty reverberation chamber 3 and the same chamber equipped with stationary and randomly hanging reflective panels. Chamber arrangement: 0, the empty chamber reference case; 1, six $1.25 \times 1.5 \text{ m}^2$ panels, each weighing 6.7 kg/m^2 , hanging in the room (relative panel area: 14 %).

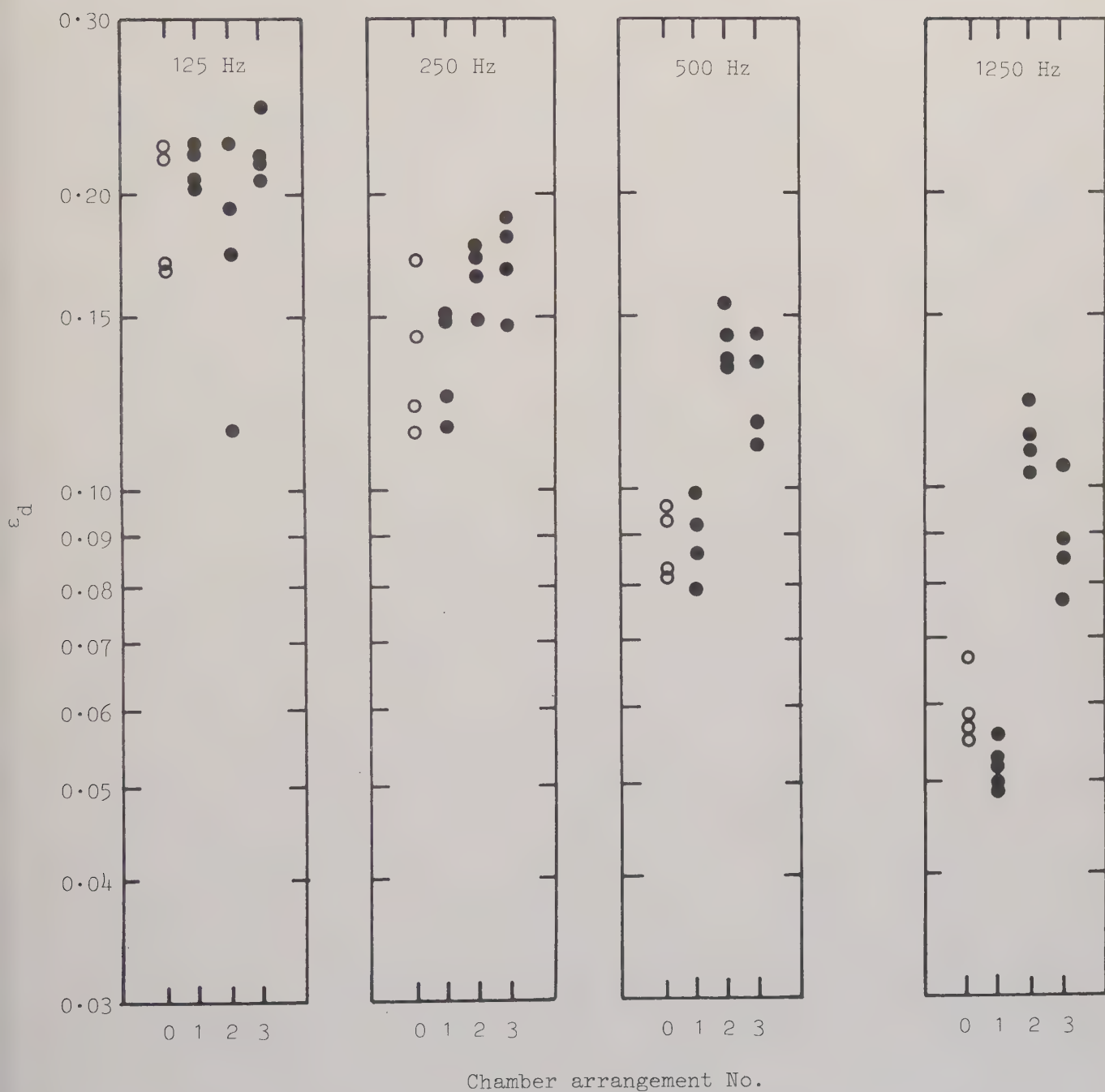


FIGURE 12. Comparison between measured diffusivity values for the empty reverberation chamber 2 and the same chamber equipped with stationary hanging panels or/and with absorptive patches randomly placed at the boundaries. Chamber arrangement: 0, the empty chamber; 1, 10 reflective panels, each weighing 6.7 kg/m^2 , hanging in the room (relative panel area: 28 %); 2, 10 absorptive patches added to the boundaries (total area of porous absorbents $\approx 9 \text{ m}^2$); 3, chamber arrangements 1 and 2 combined.

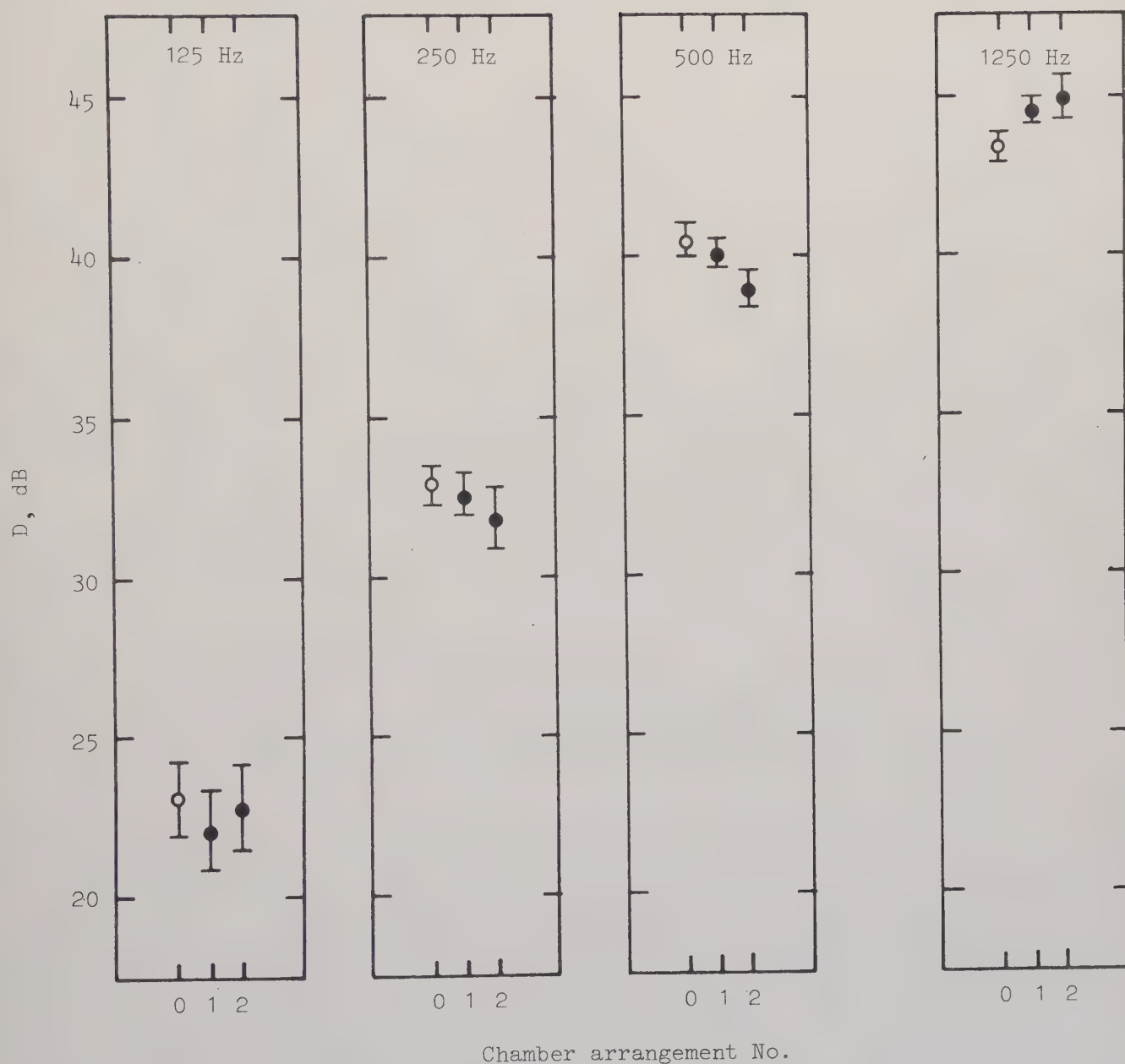


FIGURE 13. Comparison between sound pressure level difference values obtained in the transmission loss laboratory when measuring on the same window as in Figure 7 and when three different source chamber arrangements are used. Chamber arrangement: 0, the empty chamber reference case; 1, six $1.25 \times 1.5 \text{ m}^2$ panels hanging in the chamber (relative panel area: 14 %); 2, arrangement No. 1 and 10 absorptive patches added to the chamber boundaries.

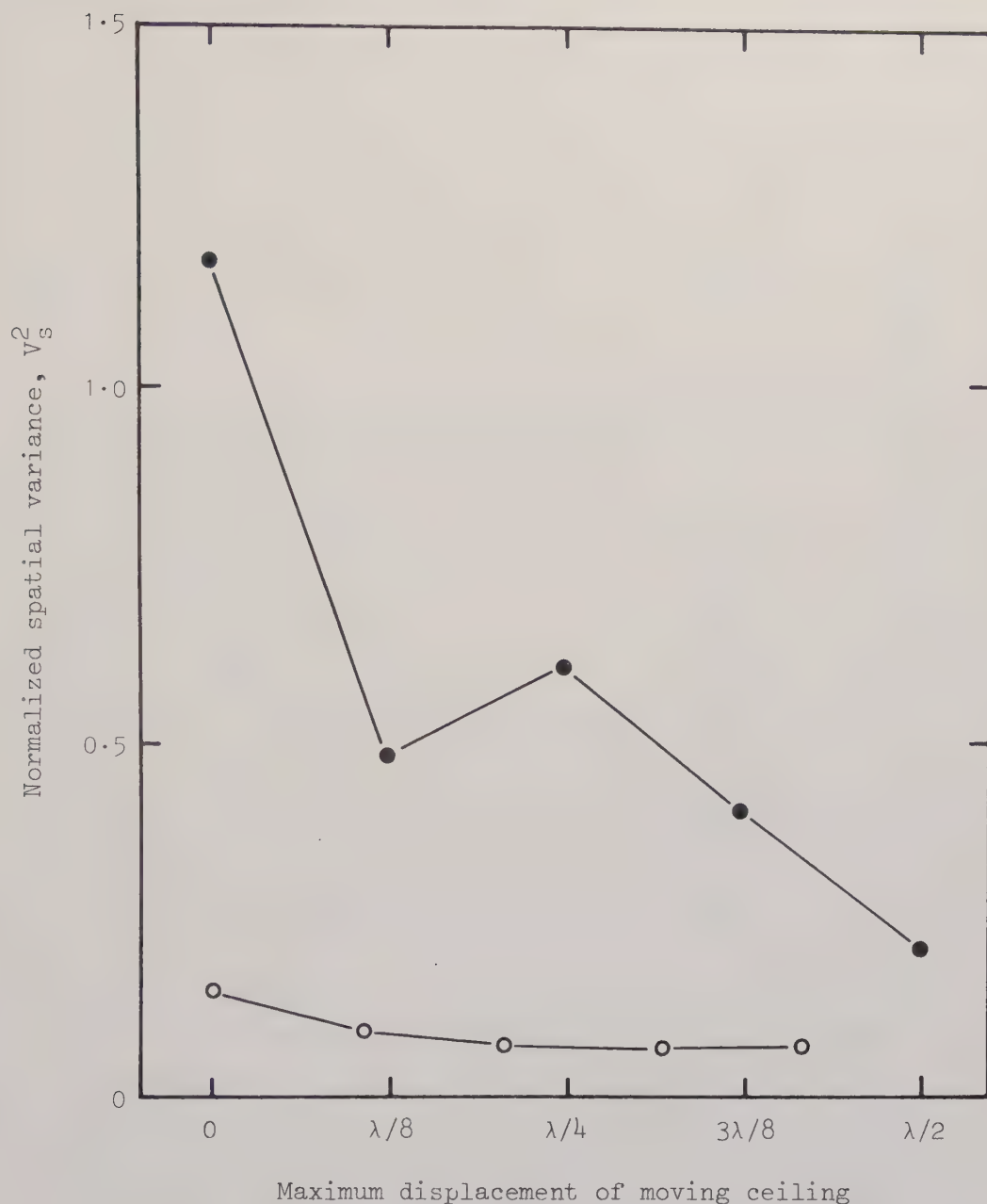


FIGURE 14. Experimental values of the normalized spatial variance V_s^2 obtained when the model reverberation chamber was equipped with a vertically and continuously moving ceiling. This piston was made to move up and down in simple linear motion, and the maximum displacement from the top position has been varied. The chamber was excited with a 1250 Hz pure tone (●) and with a 1250 Hz 1/3-octave band noise (○). To estimate a variance value 50 random discrete spatial mean square pressure samples were taken.

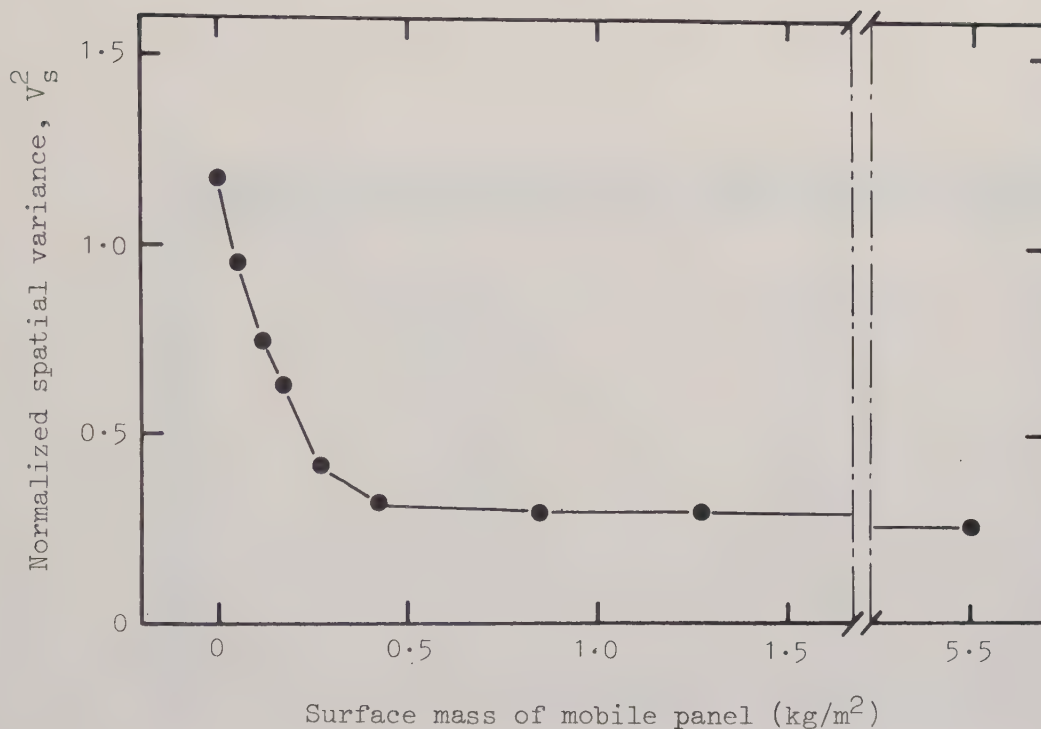


FIGURE 15. Experimental values (●) of the normalized spatial variance obtained in the model reverberation chamber. The chamber was equipped with a vertically and continuously moving ceiling, the surface mass of which has been varied. This piston was made to move up and down in simple linear motion, and a maximum displacement of 0.135 m ($\lambda/2$) was used. The chamber was excited with a 1250 Hz pure tone, and to estimate a variance value, at least 50 random discrete spatial mean square pressure samples were taken.

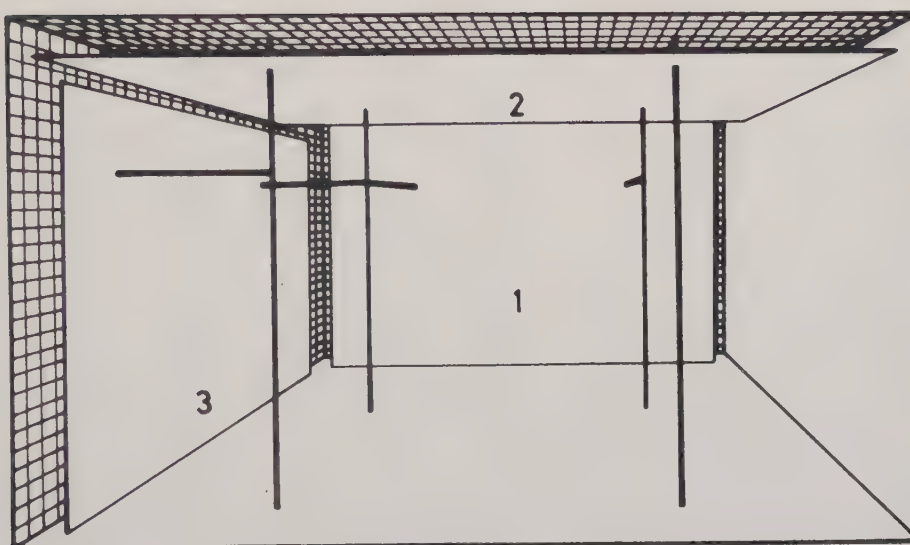


FIGURE 16. A sketch of the moveable boundary system built up in the source room (chamber 3) as seen from the receiving side of the transmission loss laboratory. The area of reflector No. 1 is 90 % of the area of the chamber wall behind it, and the reflectors Nos. 2 and 3 are taking up 73 % and 62 % of the respective wall areas. The test frame between the source and the receiving room has been omitted in the figure.

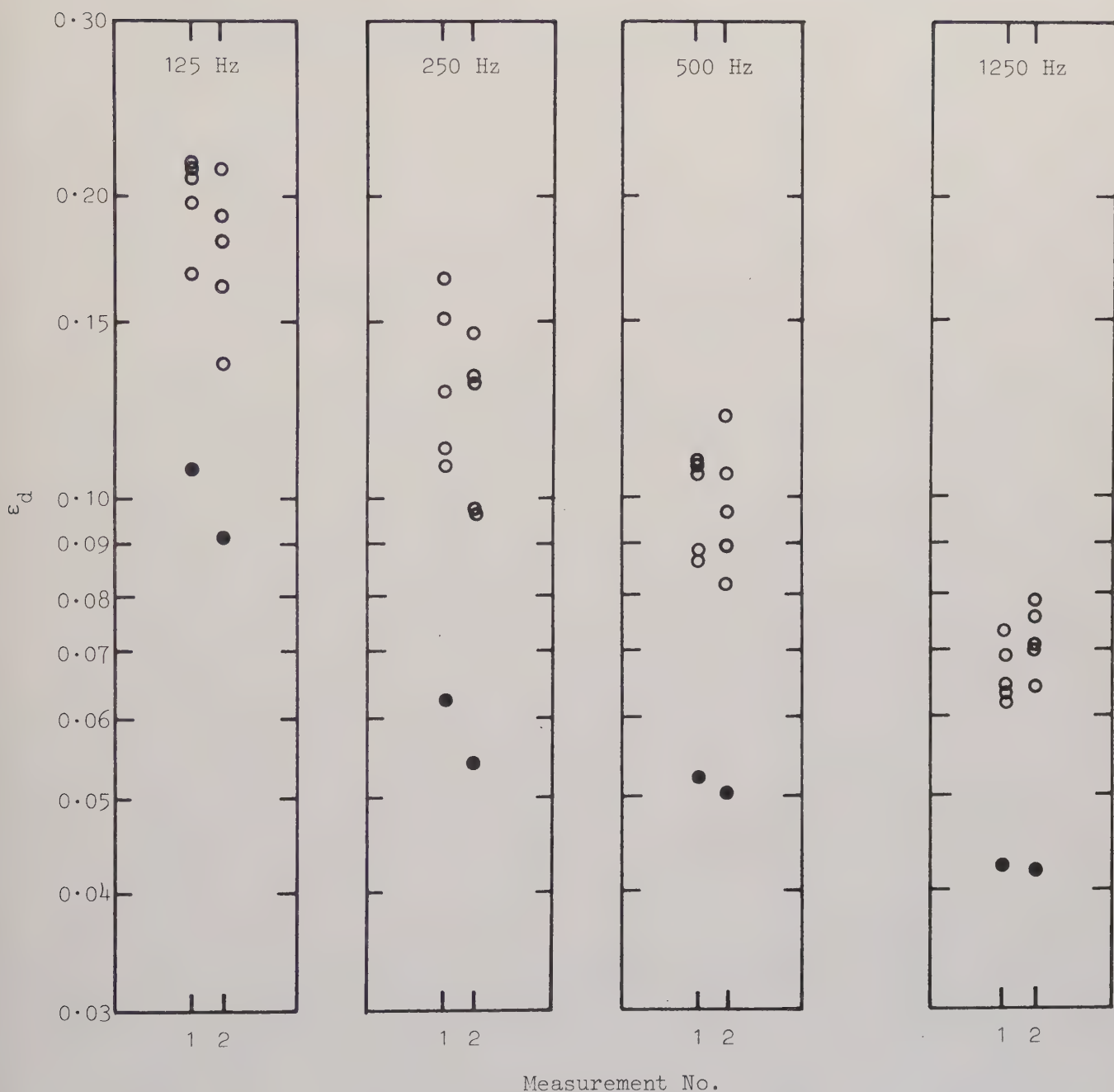


FIGURE 17. Measured diffusivity values obtained when the reverberation chamber 3 was equipped with the moveable boundary system. The room averaging procedure with five specific discrete boundary configurations was studied. The values obtained for each one of the five boundary configurations (o) and the respective room averaged values (●) are presented for two successive measurements.

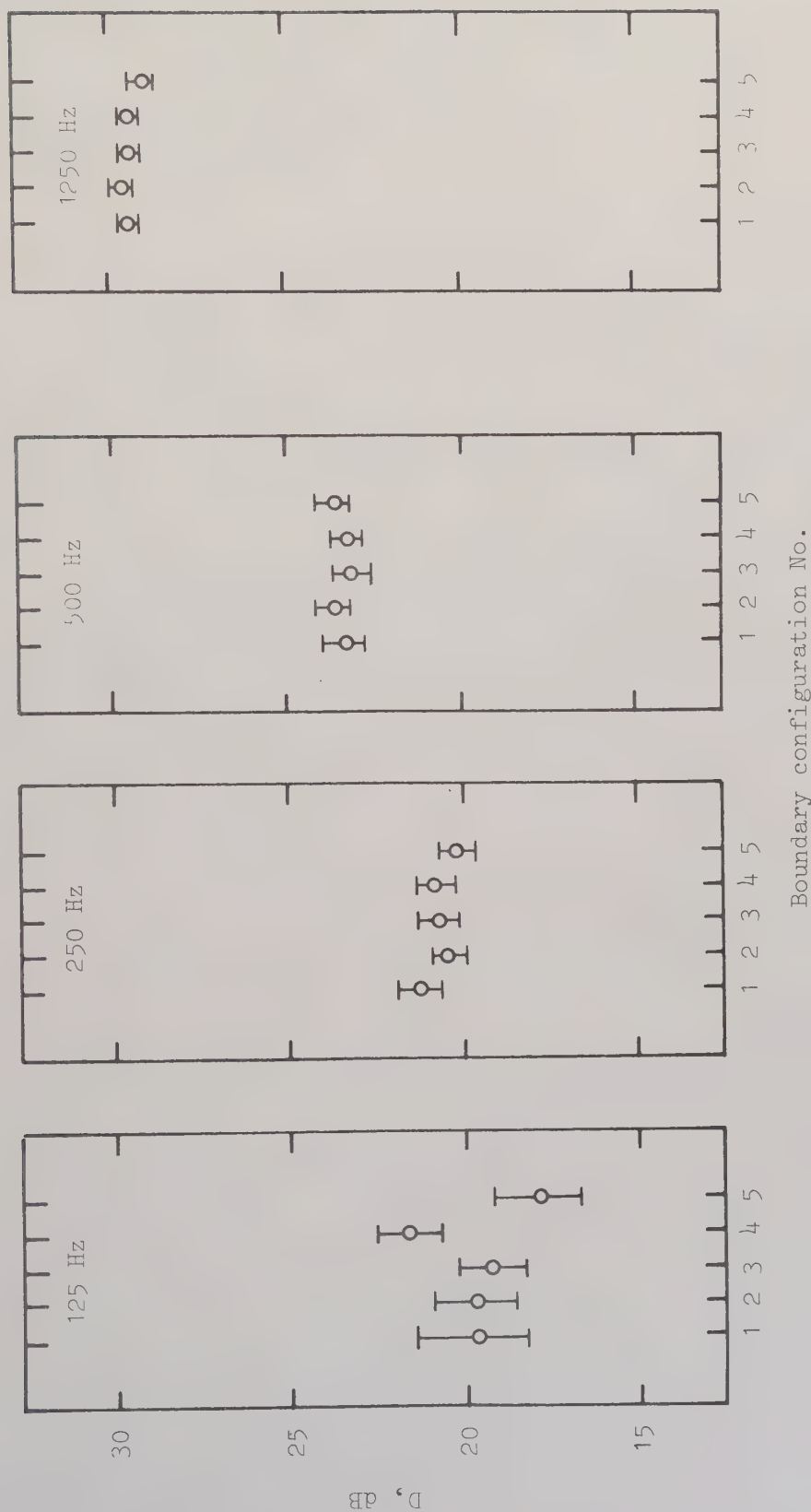


FIGURE 18. Comparison between sound pressure level difference values obtained when the source room of the transmission loss laboratory was equipped with the moveable boundary system. The values obtained for each one of the five boundary configurations are presented. A 13 mm thick, and 10.4 kg/m^2 heavy gypsum board was mounted in the test opening, flush with the frame wall of the source room.

TABLE 1

Comparison between obtained ϵ_d -values and the associated rank numbers for the reference source and the same source but without back panel and absorptive lining. The measurements were performed at 125 Hz in chamber 1.

Reference source		Open-back source	
ϵ_d	rank No.	ϵ_d	rank No.
0.219	1	0.177	7
0.208	2	0.170	9
0.188	3	0.170	10
0.185	4	0.165	11
0.182	5	0.159	13
0.179	6	0.148	14
0.171	8	0.144	15
0.160	12	0.142	16
$\Sigma R = 95$			

TABLE 2

The approximate reverberation times and the respective typical Δ -values ($r = 4.0$ m) for the reverberation chamber 3 with and without absorbents according to Figure 8.

	Frequency (Hz)			
	125	250	500	1250
T_{60} - "empty room" (s)	5.6	5.7	5.0	4.1
T_{60} - "random absorption" (s)	2.1	1.5	1.4	1.4
T_{60} - "concentrated absorption" (s)	1.9	1.6	1.5	1.3
Δ - "empty room"	0.04	0.04	0.04	0.05
Δ - "random absorption"	0.10	0.14	0.15	0.15
Δ - "concentrated absorption"	0.11	0.13	0.14	0.16

TABLE 3

Experimental values of the normalized spatial variance V_s^2 obtained when the model reverberation chamber was equipped with various rotating diffusers and when the chamber was excited with a 1250 Hz pure tone. To estimate a variance value 50 random discrete spatial mean square pressure samples were taken.

No.	Description of diffuser			V_s^2
	Type	One-sided area, m^2 *	Further details	
0	No diffuser	0		1.17
1	Quadrate	$\lambda^2/16$	Axis coinciding with midline	0.91
2	Quadrate	$\lambda^2/8$	Axis coinciding with midline	0.78
3	Quadrate	$\lambda^2/4$	Axis coinciding with midline	0.72
4	Quadrate	$\lambda^2/2$	Axis coinciding with midline	0.57
5	Quadrate	λ^2	Axis coinciding with midline	0.38
6	Quadrate	$1.25 \lambda^2$	Axis coinciding with midline	0.42
7	Quadrate	$1.5 \lambda^2$	Axis coinciding with midline	0.41
8	Quadrate	λ^2	Axis intersecting centre/ 30° to midline	0.67
9	Two rectangular vanes	λ^2	Vanes being $45^\circ/180^\circ$ to each other	0.49
10	Semicylindrical shell	λ^2	Axis coinciding with midline of shell	0.68
11	Truncated pyramid	$1.08 \lambda^2$	Two adjacent sides covered	0.72
12	Truncated pyramid	$1.08 \lambda^2$	Two opposite sides covered	0.43
13	No.4 + No.5	$1.5 \lambda^2$		0.33
14	No.12 + No.5	$2.08 \lambda^2$		0.35

* λ is the wavelength of sound in air at 1250 Hz and $20^\circ C$

TABLE 4

Geometrical characteristics of the moveable boundary system
expressed in the wavelength (λ) of sound in air at 125 Hz and
20° C.

	Boundary		
	No. 1	No. 2	No. 3
One-sided area	$2.3 \lambda^2$	$3.7 \lambda^2$	$1.9 \lambda^2$
Maximum possible displacement	0.50λ	0.12λ	0.27λ

